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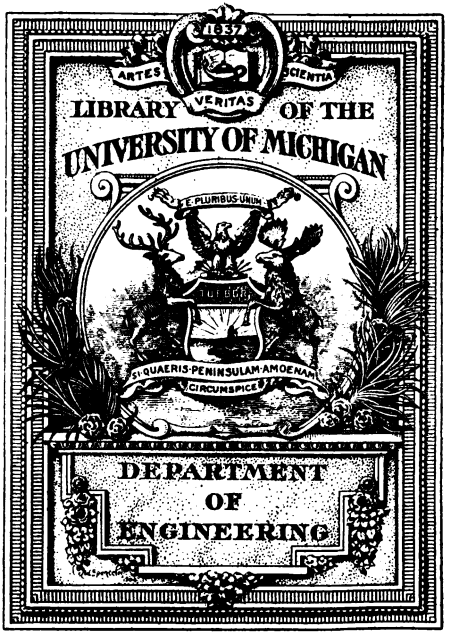
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SUPPLEMENT

TO THE

STEAM ENGINE CATECHISM.

SUPPLEMENT

TO THE

33829

STEAM ENGINE CATECHISM:

A SERIES OF

DIRECT PRACTICAL ANSWERS TO DIRECT PRACTICAL QUESTIONS

MAINLY INTENDED FOR THE USE OF

YOUNG ENGINEERS AND FOR EXAMINATION PURPOSES.

BY

ROBERT GRIMSHAW, M.E., ETC.,

PAST PRESIDENT JAMES WATT ASSOCIATION No.7, N.A.S.E.,

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THIRD AND ENLARGED EDITION.

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TO THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS

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TO THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS

PREFACE
TO FIRST EDITION OF PART II.

The matter in this volume is intended as supplementary to the Catechism, and needs no excuse for its appearance.

The extreme popularity and rapid sale of the Catechism proper warrants the author in putting in the same form, additional material of the same general character.

This volume contains the answers to many questions which were asked by engineers and steam users, and the answers to which do not appear in the Catechism itself.

It refers in many places to the Catechism, and is intended to be used with it.

Particular attention is paid to compound engines, to link motions, to the details of various existing types of steam engines, and to matters concerning setting up, running, adjusting, repairing, and altering particular builds of engines.

There is a probability that another supplementary volume may be issued in the course of a short time.

PREFACE
TO THIRD EDITION OF PART II.

In this edition the reader will find about 22 pages added, embracing about 70 questions and answers, mostly concerning the slide valve; while a number of typographical errors (none of them, however, important as affecting the sense) have been corrected. The Index, covering nineteen closely printed pages, will be found to contain the proper references to the new matter just added.

If there is any question relating to any matter concerning any kind of a steam engine, and which is not answered either in Part I. or in this volume, I should be very happy to receive it for reply by mail; and such questions and their answers will probably be added to future editions or volumes.

5 Dey St., N. Y., Nov. 1, 1887.

SUPPLEMENT

TO THE

STEAM ENGINE CATECHISM.

EXPANSION, &C.

Q. What is the advantage of an adjustable cut-off on marine engines?

A. There are times where the engineer can take advantage of smooth seas and favorable winds, and use less steam because the work is less.

Q. Is there any advantage of early cut-off on locomotives?

A. Mr. H. S. Hayward, Master Mechanic of the N. Y. Division of the Penn. R.R., says in a paper before the Am. Soc. Mech. Eng.:—

"In my railroad experience I find that a great deal of economy has also been realized in our locomotives by encouraging engineers to carry their steam up to the highest point allowed them, and cutting off their engines shorter instead of using the throttle."

Q. What is the disadvantage of very early cut-off on marine engines?

A. Where the cut-off is too early the engines are apt to race, and the wheel velocity to vary greatly.

Q. How may this be remedied?

A. This may be remedied in part by making the outer rim of paddle wheels very heavy. Mr. H. S. Hayward says, speaking of the ferry boat *Baltimore*, where the cut-off is adjustable and at times very early:—"We turn a very even wheel. We have constructed the outer ring of our wheel very much thicker than heretofore, and the boat is more free from the vibrations which you find on ferry boats."

Q. What is the disadvantage of too early cut-off on locomotives?

A. Wheel slippage, in many cases.

Q. On vertical engines should the cut-off be later on the upward or on the downward stroke?

A. On the down stroke.

Q. How is clearance volume often reckoned?

A. In parts of the stroke supposed to be multiplied by the piston area. Thus, if the

piston stroke is 20″ and area 200 square inches, and the clearance volume at one end is 200 cubic inches, then the clearance would be reckoned as equal to 200÷200=1″ of stroke.

Q. Supposing expansion perfect, under proper conditions, how may the "absolute" steam pressure in the cylinder, after cut-off, be ascertained?

A. Supposing expansion perfect, under proper conditions, the "absolute" steam pressure in the cylinder, after cut-off, should be just proportionate to the cylindrical volume bounded by the cylinder head at one end and the piston head at the other.

Assuming for the present that there is absolutely no clearance space between the piston head and the cylinder head when at either stroke end, and that cut-off takes place at one fourth stroke or 6″ in an engine having 24″ stroke, the terminal pressure above vacuum will be one fourth that at the point of cut-off; the pressure (*above vacuum*) when the piston has made half a stroke, or 12″ advance, will be half that at the point of cut-off; when it has made three fourths stroke, or 18″ advance, it will be one

third that at the cut-off, and so on. In other words, *the pressure above vacuum is inversely proportioned to the volume the steam occupies.*

Q. With cut-off at $\frac{1}{4}$, what initial pressure would be needed to get a mean effective pressure of ten (10) lbs., in a non-condensing engine, supposing there to be no clearance, cushion, wire drawing, throttling, leak, cylinder condensation, etc.

A. The average total pressure required would be $10+14.7=24.7$ lbs.; the expansion rate 4, which has 1.3863 for its hyperbolic logarithm; then the total initial pressure above vacuum would be $24.7\div(.25\times1+1.3863)=24.7\div(.25\times2.3863)=24.7\div.59658=41.4$ lbs.; and the initial pressure by the gauge would be $41.4-14.7=26.7$ lbs.

Q. Suppose the foregoing conditions to be changed by a back pressure of 2 lbs., what initial pressure would be required?

A. Then the average total pressure required would be $10+2+14.7=26.7$ lbs., and the initial total pressure $26.7\div.59658=44.7$ lbs.; hence the initial pressure "by the gauge" would be $44.7-14.7=30$ lbs.

Q. At what point in the stroke would

the cut-off have to take place, in the above mentioned case, to make the average effective pressure 30 lbs., instead of 10?

A. As the mean effective pressure can never be greater than the initial gauge pressure by the gauge in a non-condensing engine, and as the desired mean effective pressure is 30 and the initial gauge pressure only 26.7 lbs., the engine couldn't do the work.

Q. How could the work be done, then?

A. By adding a condenser.

Q. At what point in the stroke would the engine have to cut off-to give 30 lbs. mean effective pressure, with 26.7 lbs. initial pressure by the gauge, using a condenser?

A. Depends on the amount of vacuum. If the condenser gave a vacuum of only 5 lbs., then the engine would have to give 25 lbs. average pressure above atmosphere (39.7 lbs. av. total pressure) besides the work done by the condenser; and as the initial pressure is only 26.7 lbs., cut-off would have to be very late. In figures, — as the hyperbolic logarithm of the actual expansion rate is found only by dividing the

average total by the initial total pressure, and the quotient by the period of full steam plus the clearance, and then taking one from the quotient, we have two unknown quantities to work with, and the calculation will have to be made by approximation, or on the "cut and try" principle.

Assume cut-off at $\frac{9}{10}$, (expansion rate 1.111, the hyperbolic logarithm of which is 0.1044) we find that to give us an average total pressure, (with 41.4 initial total), of

$$41.4\ (0.9 \times 1.1044) =$$
$$41.4 \times .99396 = 41.14 \text{ lbs.},$$

which is too much.

Trying cut-off at $\frac{8}{10}$ (expansion rate 1.25, with hyperbolic logarithm .2231) we have that giving an average total pressure (with 41.4 initial) of

$$41.4\ (0.8 \times 1.2231) =$$
$$41.4 \times .97848 = 40.5$$

Trying cut-off at $\frac{7}{10}$ (actual expansion rate about 1.43, with hyperbolic logarithm 0.3577); that would give, with initial total of 41.4 lbs.,

$$41.4\ (0.7 \times 1.3577) =$$
$$41.4 \times .95039 = 39.35 \text{ lbs.},$$

which is less than the 39.7 which we want;

so we know that the cut-off required for 5 lbs. vacuum will be between 0.7 and 0.8 stroke.

For other amounts of vacuum the principle is the same.

Q. How is the work of the steam during admission (including overcoming atmospheric and other counter pressure) reckoned?

A. By multiplying the total pressure on the piston by the period of admission.

Q. How is the volume of steam admitted or used during the "full steam" period, measured?

A. It is equal to the product of the piston area by the period of admission plus the clearance.

Thus, if there is 2″ of clearance and cut-off takes place at 5″ of stroke, and the piston area is 260 square inches, the volume of steam is $260\times(2+5)=260\times7=1820$ cubic inches.

Q. How do you calculate the total work done *during* expansion to the end of the stroke?

A. Multiply the total pressure above vacuum on the piston area by the period of

admission plus the clearance, and by the hyperbolic logarithm of the actual expansion rate. (This includes the work done in overcoming all counter pressure above vacuum.)

Q. How do you find the initial pressure required for a given net quantity of work in one stroke?

A. To the net work expressed in foot pounds, add the continuous product of the piston area in square inches, the average back pressure in lbs. per square inch for the whole stroke, and the length of stroke in feet; divide the sum of these quantities by the piston area in square inches, and by a number got thus:—Add one to the hyperbolic logarithm of the actual expansion rate; multiply the sum by the period of admission plus the clearance (both in feet); and take the clearance in feet from the product.

As a formula:—

$$P = \frac{W + a\,p'\,L}{a\,[\,l'\,(1 \times \mathrm{hyp.log}\,R') - c]}$$

Thus:—Suppose the given net work per single stroke to be 2,000 foot pounds; the

piston area 260 square inches; the average back pressure one pound per square inch by the gauge; the stroke 2′; cut-off at ¼ stroke, and clearance equal to 0.1 of stroke; then we have—

2,000+(260×15.7×2) = 10,164 ft. lbs; 10,-164÷260 = 39.09; the actual expansion rate 2.1÷1.1=nearly 1.91, the hyperbolic logarithm of which is .6472; (1.1×1.6471)—0.1= 1.7118; 39.09÷1.7118 =23.83 lbs. absolute; 23.83—14.7 = 9.13 lbs. by the gauge.

(The engine would develop at 110 revolutions per minute 2,000×110×2 = 440,000 ft. lbs., or 440,000÷33,000 = 13.3 H. P.)

Q. How do you find the net work done by the steam for one stroke of the piston, with a given cut-off?

A. Find the actual expansion rate. Get its hyperbolic logarithm and add one thereto. Multiply the sum by the period of admission plus the clearance (in feet). From the product take the clearance. Multiply the remainder by the total initial pressure in lbs. per square inch. This gives the *total* work done, in foot pounds per square inch, on the piston. To get the negative work per square inch of back pressure in foot

pounds per square inch, multiply the average back pressure in pounds per square inch by the length of stroke in feet. Take the product from the total work per square inch, and this will give the net work per square inch. Multiply the remainder by the piston-area; this gives the net foot pounds for one stroke.

Q. How do you find the period of admission required for a given actual expansion ratio?

A. Multiply the clearance proportion by one less than the actual expansion ratio; take this from one; divide by the actual expansion ratio.

Thus:—Suppose 2′ stroke, actual expansion rate 3.5; clearance five per cent; then the period of admission is

$$\frac{1-[(3.5-1)\times.05]}{3.5}=\frac{1-.125}{3.5}=.25 \text{ stroke,}$$

(or cut off at six inches).

Q. How do you find the pressure at stroke end, or at any other point in the stroke after expansion, of steam expanded in the cylinder?

A. Divide the initial pressure above vacuum by the actual ratio of expansion

calculated to the given point of the stroke.

Or, multiply the initial pressure by the period of admission plus the clearance, and divide the product by the length of the part of stroke described up to the given point, plus the clearance.

Thus:—Suppose initial pressure sixty lbs. by the gauge, clearance five per cent, cut-off at one-fourth, required the pressure at ¾ stroke. The expansion rate *up* to *three-fourths stroke* is $(.75+.05)\div(.25+.05)=.80\div.30=2.667$ and $(60+14.7)+2.667=74.7+2.667=28$ lbs. absolute, or $28-14.7=13.3$ lbs. by the gauge.

Or, $$\frac{74.7\times(.25+.05)}{.75+.05}=\frac{74.7\times.30}{.80}=28 \text{ lbs.}$$

above vacuum, or 13.3 lbs. by the gauge.

Q. How do you find the net cylinder capacity for a given quantity of steam admitted for one stroke and a given actual expansion rate?

A. Multiply the volume of one lb. of steam at the given pressure (as found in a reliable table) by the given weight in pounds and by the actual expansion rate. Multiply

the result by 100 and divide by 100 plus the clearance percentage.

Thus:—Suppose we want to expand five lbs. of steam at 100 lbs. per square inch above vacuum at an actual expansion rate of 3.5, in a cylinder where clearance is 5%.

The volume of one lb. of steam at 100 lbs. absolute, is 4.33 cubic feet. Thus the net capacity is

$$\frac{5 \times 4.33 \times 3.5 \times 100}{1.05} = 72.167$$

cubic feet, or, say a cylinder of about 4 stroke and eighteen square feet of piston area, or having a diameter of about $4.8' = 57.6''$.

Q. How do you find the net cylinder capacity required to do a certain amount of total actual work in one stroke, with a given initial pressure and actual expansion ratio?

A. Divide the given work by the total actual work done by one lb. of steam of the same pressure and with the same actual expansion rate; this gives the weight of steam required, and from that can be found by a preceding rule the net capacity.

Q. How do you find the weight of steam admitted per cubic foot of the net cylinder capacity, for one stroke?

A. Divide 1 by the cylinder capacity per pound of steam.

Q. How do you determine the total actual work done per cubic foot of net capacity, for one stroke?

A. Divide 1 by the cylinder capacity per foot pound of work done.

Q. How do you find the total actual work done per square inch of piston?

A. The total amount of actual work done per square inch of piston is $\frac{1}{144}$ part of the work done per square foot. (See above.)

Q. What are the causes of differences between boiler and initial cylinder pressure?

A. The causes, commencing at the boiler, are friction in the steam pipe, resistance of the regulating throttle valve, resistance in the ports and passages, and disappearance of actual energy when the steam passing from the small area of the ports into the larger area of the cylinder, has its speed reduced.

Q. How does back pressure by the gauge vary?

A. Back pressure by the gauge varies in non-condensing stationary engines and in locomotives nearly

As the square of the speed.

As the pressure of the steam at the instant of release (commencement of exhaust).

Inversely as the square of the area of the orifice of the exhaust nozzle.

It is less with high expansion rates than with low; and less with slow exhaust than rapid; less with dry than with wet steam, hence greater in unprotected than in well-covered cylinders (the proportion being 1.72 to 1 in observed cases).

It may be said for all engines, condensing or non-condensing, that "in the same engine, going at the same speed, the excess of the mean back pressure above the pressure of condensation, varies nearly as the density of the steam at the end of the expansion; and that in the same engine, with the same density of steam at the end of the forward stroke, that excess of back pressure varies nearly as the square of the speed."

Q. "Why does not the exhaust steam show a difference of temperature corresponding to the difference of power to be

obtained from steam at boiler pressure and exhaust steam?"

[The foregoing question was asked us in POWER, and was answered as follows :—]

A. The relation between the pressure on steam and its temperature is in a varying ratio, and the variation of this ratio is of a complicated character. The pressure of steam increases with the temperature, but the *rate* of increase of temperature increases as the pressure rises. Thus at 212° F. the pressure above vacuum is 14.7 lbs. per square inch, and at 213° the pressure rises 0.29 lbs.; at 247° the pressure above vacuum is 28.34 lbs. per square inch; and at 248° it has risen about ½ lb. At 432° the pressure is 350.73 and the rise in pressure per degree of temperature is 3.64 lbs., or about fourteen times as much rise of pressure per degree, as at atmospheric pressure. This relation between temperature and pressure is the same so long as the steam is in communication with unevaporated water.

THE SLIDE VALVE.

Q. What is a rule of thumb for lap of slide valve to cut off at two-thirds?

A. To cut off at two-thirds a well proportioned valve may have lap equal to one-fourth the stroke. (Chas. E. Emery.)

Q. What is the rule of thumb for inside lap of condensing engines?

A. One-half the outside lap.

Q. Which will stand the most inside lap, a condensing or a non-condensing engine?

A. The condensing engine.

Q. How may the evils of over cushion from excessive inside lap, generally be remedied in part?

A. By giving the valve more lead, so that the final compression is against the steam in the chest and not against the valve face.

Q. How should a slide valve be set so as to equalize the exhaust?

A. Put the crank on the front centre and mark coinciding centre-punch marks on cross-head and guide. Put the crank on the back centre, and centre-punch the guide again, to correspond with the cross-head mark. Move the eccentric until the valve has the lead for the forward stroke. Move the crank, in the direction the engine is to run, until the exhaust of the opposite

stroke closes; scribe a line on the guide to correspond with the cross-head mark; move the crank on until the other exhaust closes; scribe on the guide to correspond with the cross-head mark. If then the exhaust does not close at equal distances from the stroke ends, alter the length of the eccentric rod until it does; then bring the crank back to the dead centre, and shift the eccentric until the forward stroke has the required lead.

Q. How may you tell where the valve is at the time of closure, without opening the chest?

A. By a valve gauge of stout wire rod of this shape,

fitting centre punch-marks on the valve stem and on the stuffing-box (*not* on the gland).

Q. How can you tell when the valve has got lead, without opening the chest, if there are no punch marks to go by?

A. Put the crank on the centre, open the drip cock at that end, and then turn the eccentric ahead until steam shows at the drip cock.

Q. How do you find the correct length of the eccentric rod?

A. Mark the centre lines of cylinder, valve, rocker and main shaft. Put the crank on the back centre, and the valve so that the required lead is given for the forward stroke. Find the position of the eccentric centre by laying off the angular advance from the 90° line. Measure from this centre to the rocker pin (or to the valve stem pin if there is no rocker) for correct length of eccentric rod.

Q. If the lead opening is measured correctly for the forward stroke, will it be correct for the return stroke, if the eccentric rod length is measured as above directed?

A. Yes; because the opening is measured while the crank is on the central line—hence there is no irregularity of piston position owing to the angularity of the connecting rod.

Q. Is equalization of exhaust closure always possible?

A. Yes; because there may be added a slight amount of inside lap to whichever end of the valve needs it, so as to hasten exhaust closure on one stroke and delay it on the other.

Q. In equalizing cut-off should equalization of exhaust closure be considered, also?

A. No; because the only practical method for equalizing cut-off also equalizes the exhaust well enough.

Q. On what does the expediency of equalizing cut-off depend?

A. Upon that of giving unequal lead.

Q. Where is unequal lead disadvantageous?

A. For high speed engines.

Q. What alterations in the valve motion are requisite, in order to equalize cut-off?

A. Increase in angular advance, and lengthening the eccentric rod.

Q. In what class of engines is the steam pressure in the two ends more apt to differ; in vertical or in horizontal?

A. In vertical engines.

Q. Which end of a vertical engine generally gets the highest average pressure?

A. The lower.

Q. To reverse an engine having a single eccentric, what is necessary?

A. To turn the eccentric so that its centre line shall come on the other side of the centre line of the engine.

Q. In order to keep the lead opening and the lap the same for different positions of the eccentric for a reversing engine, what is necessary?

A. That the eccentric be slotted so that its centre can pass across the centre line of the engine in a direct straight line, instead of in an arc, as where the eccentric is simply turned on the shaft.

Q. How does the position of an adjustable eccentric for a reversing engine affect the cut-off?

A. The nearer mid-gear the earlier the cut-off, whether the eccentric centre be moved along a straight line (as where the eccentric is slotted) or in an arc, (as where it is simply turned on the shaft.)

Q. How does the position of the eccentric of a reversing engine affect the exhaust?

A. The nearer mid-gear the earlier the exhaust, whether the eccentric centre move in a straight line or in an arc.

Q. How does the position of the eccentric centre in a reversing engine affect the lead angle and the angular advance?

A. The nearer mid-gear, the greater the lead angle and angular advance, whether

the eccentric centre be moved in a straight line or in an arc.

Q. What is the lead opening at mid-gear?

A. At mid-gear the lead opening of the valve equals the port opening.

Q. What is the cut-off angle at mid-gear?

A. At mid-gear the cut-off angle equals the lead angle of the valve.

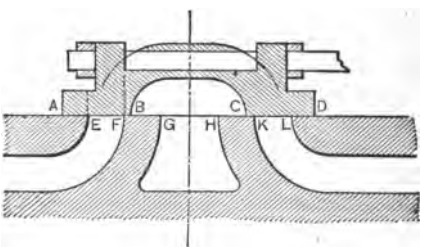
Q. How should valve lead be stated or reckoned?

A. Lead should be stated in degrees rather than in inches of either valve travel or piston motion.

Q. How should steam ports be proportioned?

A. After you have settled the throw of the eccentric, the angle of advance or angular lead, and the steam port width $EF=KL$, we might say as a general rule that the bridges FG, HK, should be wider than the distance at which the outer edge A of the valve, at its greatest distance from the central position, stands beyond the inner edge F of the admission port; otherwise the valve would blow through. The sum of the two

laps *AE* and *BF*, plus the port width *ES*, must be greater than half the valve travel; hence the bridge-width *EF* must be greater than half the valve travel, less the sum of *AE* and *BF*. You might say that the bridge-width *FG* should be equal to half the steam port width *EF* plus $\frac{1}{32}$" to $\frac{1}{16}$"; and the exhaust port *GH*, equal to half the valve



travel plus the steam port width *EF*, plus the inside lap *BF*, less the bridge-width *FG*.

Q. In designing an Allen valve for an old seat, what is generally a good thing to do?

A. To widen the end ports in the seat by chamfering off the outside edges.

Q. What precaution must be taken in designing an Allen valve for an old seat?

A. Not to let the supplementary passage in the valve get over the end port in the seat.

Q. Will the Allen valve save fuel?

A. Sinclair says, "In very carefully conducted experiments made on the Boston and Albany R. R., to compare the performance of the Allen valve with an engine equipped with a common valve, it was found that the Allen valve effected a fuel saving of seven per cent."

Q. What difference should there be made in the seats in designing an Allen valve and an ordinary slide?

A. For the Allen, the steam ports and bridges should be a trifle wider than for the ordinary slide.

Q. What special care should be taken in casting an Allen valve?

A. To get the supplementary passage smooth inside, and the thin shell strong enough to stand the pressure.

Q. When is the speed of a slide valve highest?

A. The speed of a slide valve is greatest at mid-throw.

Q. How is the D valve kept to its seat?

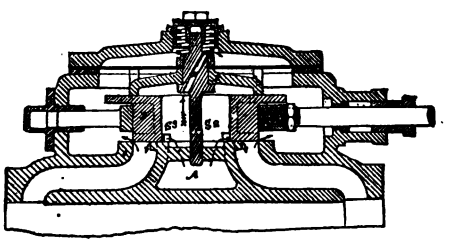
A. The D valve is kept to its seat by the

excess of the boiler pressure over that below the valve. The trouble is that it is too strongly held to its seat, causing undue friction, with its accompanying abrasion, and loss of power.

Q. What is one disadvantage of most balanced slide valves?

A. They do not lift if there is water in the cylinder.

Q. What is Tipping's balanced slide valve?



A. In this valve the steam is admitted at *A* inside the slide valve *S*, and is exhausted through the steam chest *C*; the valve *S* is open through, *B* is a back plate resting thereon nearly in equilibrium, *E* is a diaphragm plate, *F* are nuts, tightening which will relieve the pressure on the valve, *G* is a

spiral spring, the compression of which will add pressure in adjusting the valve. This arrangement affords a light, handy, and simple device, which we understand works very well, and may be constructed either rectangular or circular in shape and to any size, and for either single or double ports. In the cut the valve stem is not carried through the cover.

VALVE MOTIONS.

Q. How may valve motions be classified?

A. Valve motions may best be divided into those with fixed, and those with variable cut-offs; and the latter may have the cut-off variable either only while the engine is stationary, or while it is running. The latter, again, may be affected either "by hand" only, or "automatically." (This last is a point neglected by Zeuner in his celebrated standard treatise on valve-gears.)

Q. What is the Dodd motion?

A. The Dodd motion (1839) moves the eccentric centre directly ascross its shaft by two wedges with reversed points. It was improved by Dubs of Glasgow.

Q. What are the objections to the Dodd wedge motion?

A. It cannot be corrected for inequalities of cut-off and exhaust closure without introducing pernicious inequality of port opening at mid-gear.

Q. What is the link operated by two fixed eccentrics intended to imitate, in its effects?

A. A single movable eccentric.

Q. When and by whom were two fixed eccentrics first used for reversible gear?

A. In 1829, by Wm. T. James of New York, on a steam carriage.

Q. How many classes of link motions are there?

A. Four—

1. Shifting link.
2. Stationary link.
3. Allan.
4. Walschäert.

Q. Who invented the so called Stephenson shifting link, and when?

A. Howe, in 1843.

Q. Who invented the stationary link, and when?

A. Daniel Gooch, about 1845.

Q. Who invented the so-called Walschäert motion?

A. Heusinger von Waldegg.

Q, What are the advantages of the link and two fixed eccentrics over a single shiftable eccentric?

A. It can be adjusted for irregularities in cut-off and exhaust closure due to the angularity of the connecting rod.

Q. How can the link be suspended so as to equalize the cut-off?

A. So as to give it, on the forward stroke of a direct acting engine, more elevation, so as to neutralize the tendency of the connecting rod and to cause later cut-off on the forward stroke.

Q. What is the disadvantage of suspension so as to equalize the cut-off?

A. There is caused a certain amount of reciprocating *slip* of the link on the block at each revolution, causing lost motion and wear.

Q. In marine engines which is the more important—equalization of the cut-off or reduction of slip?

A. Reduction of slip.

Q. Where is slip least?

A. Slip is the least when the link block is nearest the point of suspension.

Q. What does this indicate?

A. That where you need a minimum slip at a certain point of suspension, the saddle stud should be placed as nearly as possible *over* such point.

Q. With a stationary link, does the lead change with changes in gear position?

A. No.

Q. Are there any shifting link motions in which the lead remains the same at all gears in both directions?

A. No, but many shifting link motions can be arranged so as to give constant lead for all gears *in one direction.*

Q. Where must the tumbling shaft be placed?

A. Either above or below the central line of motion, so that either eccentric rod shall strike it when the gear is shifted from full forward to full backward gear.

Q. Should the eccentric rod be curved?

A. No.

Q. What should be the hanger length?

A. The hanger should be long enough

not to conflict with the tumbling shaft arm in any gear. The latter should be as long as, or longer than the hanger.

Q. Suppose the boiler or other obstacle prevents the link being put in full gear back, what must be done?

A. Either the tumbling shaft must be put below the link motion or the rocker arm lengthened so as to depress the central line of the motion and the whole motion. In this case the rocker arm centre should be at an inclination so that the one driving the valve stem should be at right angles thereto, and the other one at right angles to the central line of motion.

Q. What is the influence of rocker arm length on the valve stem?

A. The longer the rocker arm, the less the vibration of the valve stem.

Q. How is the slip of the link-block affected by the rocker-arm length?

A. The longer the rocker-arm the less the slip of the link-block.

Q. Can the link motion be laid out at an inclination to the piston motion without affecting the link action?

A. Yes; so that the angular advance of

the eccentric is laid off from a line at right angles to the central line of the link motion.

Q. With "open" or "uncrossed" locomotive eccentric rods, how are the rods to be attached?

A. The "forward" rod with the top and the "backward" rod with the bottom of the link.

Q. With "crossed" eccentric rod on a locomotive, how should the eccentric rods be attached?

A. The "forward" rod to the bottom and the "backward" rod to the top of the link.

Q. How might the inequality of compression due to the eccentric rod obliquity be overcome?

A. The compression might be equalized by varying the inside lap of one end.

Q. When the valve has equal lap on both sides how may the cut-off be equalized?

A. Where the lap is equal on both sides the cut-off may be equalized by lengthening the valve stem, which has the same effect as lengthening the laps for the forward valve-stroke, and *vice versâ*.

Q. How can the inequality of the cut-

off which would be caused by eccentric rod obliquity, be neutralized?

A. Unequal cut-off by reason of eccentric rod obliquity may be neutralized by giving the valve different steam lap on the two ends; but this would give unequal steam lead.

Q. How can the effect of the obliquity of the eccentric rod be neutralized?

A. The effect of the obliquity of the eccentric rod can be neutralized by lengthening the valve stem, so as to cause the greatest port opening at fore-stroke to be rather less than on the return; setting the valve "by the lead," that is, with the eccentric rod in its inclined position.

Q. When does the obliquity of the connecting rod most affect the position of the piston?

A. The obliquity of the connecting rod affects the position of the piston most when the crank is at right angles.

Q. At mid-gear what is the angular advance of the valve?

A. Ninety degrees (90°) of arc.

Q. At mid-gear where does the exhaust close?

A. At mid-gear it closes at half stroke.

Q. Under what circumstances is steam not admitted to the cylinder for either stroke, at mid-gear?

A. When the port opening is 0 at mid gear. (Then the lead opening is 0 and the lead angle 90°.)

Q. Is port opening reduced or increased towards mid-gear?

A. Decreased.

Q. Does the lead angle increase or decrease from full gear to mid-gear?

A. It increases, whether the lead *opening* remains constant or varies.

Q. With the Stephenson link motion and uncrossed eccentric rods, where is the lead least?

A. With the Stephenson link motion and uncrossed eccentric rods, the lead is least at full stroke.

Q. With the Stephenson link motion and uncrossed eccentric rods, where is the port opening least?

A. At mid-gear.

Q. With a "crossed rod" link motion, where is the lead least?

A. At mid-gear.

Q. With a "crossed rod" link motion, is the port opening less or greater at intermediate gear, than with the uncrossed rods?

A. Less than with the uncrossed rods.

Q. What should be the length of the link?

A. The link slot may have an available length of six times the throw of the eccentrics.

Q. What causes affect the variation of the lead at different grades with Stephenson link motion?

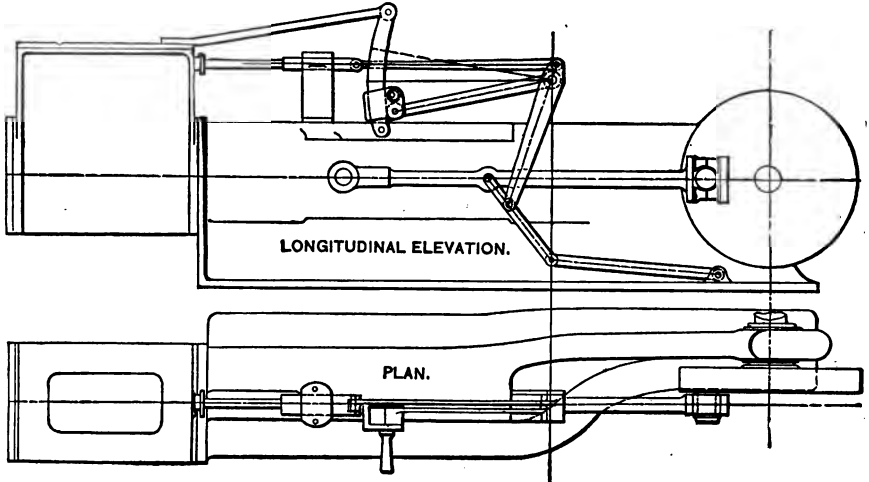
A. The variation of the lead at different grades, with the Stephenson link motion, is greatest with short eccentric rods and with long links.

Q. When a loose eccentric has lead, how far apart are its positions of forward and backward gear?

A. When a loose eccentric has lead, its positions of forward and backward gear have an angular distance equal to 180° less twice the angle of lead.

Q. When a loose eccentric has no lead, how far apart are its positions of forward and backward gear?

A. When a loose eccentric has no lead,



JOY VALVE GEAR.. SWORD ARM DESIGN.

its forward and backward gear positions are 180° apart.

Q. What is the Marshall gear?

A. In the Marshall gear, the eccentric rod bears or constitutes a link, its free end traveling in a slot in a disc which may be changed in its position. The valve stem is drawn at right angles to the centre line of the eccentric rod or link.

Q. What is the Joy gear?

A. The Joy gear drives from the connecting rod, using no eccentric. We give illustrations of three different designs of this gear.

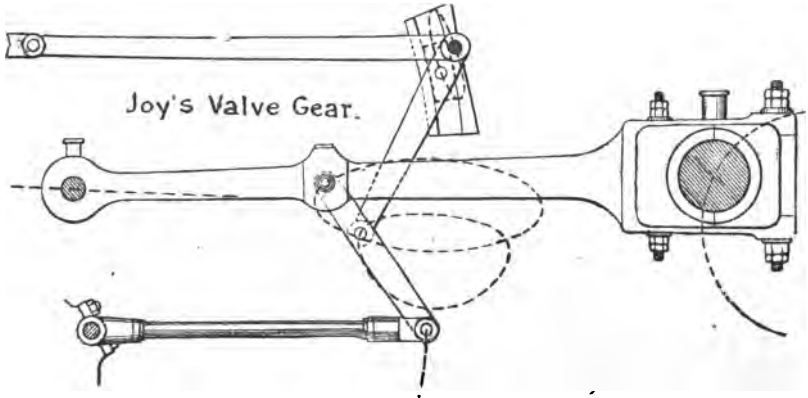
Q. What is the Brown gear?

A. The gear of Chas. Brown of Winterthur, dispenses with the eccentric, and the link is attached to the connecting rod.

THE COMPOUND ENGINE.

Q. What is a compound engine?

A. A compound engine is one in which, instead of the steam completing its expansion in the same cylinder in which it receives it from the boiler, it is only partly expanded in that cylinder, and the expansionis continued or completed in one or more cylinders (called low pressure cylinders) which



SLIDING QUADRANT DESIGN.

receive the exhaust of the first (or high pressure) cylinder. Sometimes one high pressure serves equally two low pressure cylinders in which the expansion commenced in the first is continued to an equal degree one with the other. Sometimes the expansion begun in one cylinder is continued in the second and completed in a third.

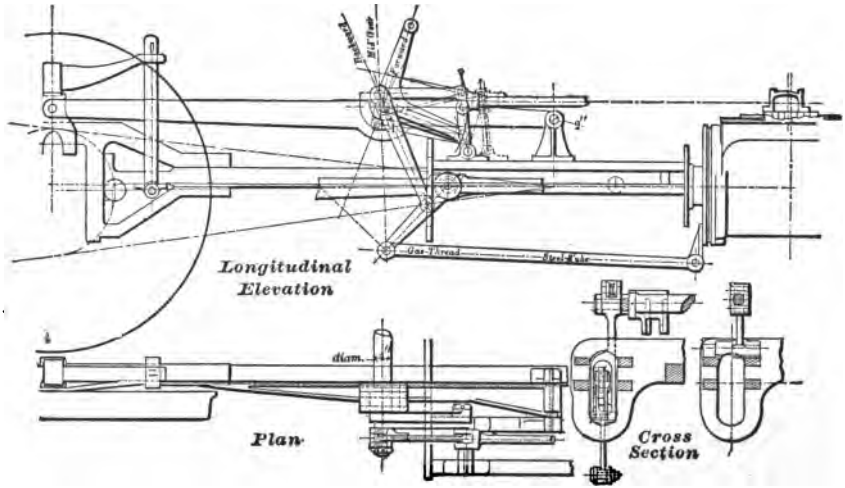
Q. For a compound condensing engine to exert the same power as a single cylinder condensing engine with the same number of expansions in each case, what would have to be the diameter of the condensing or low pressure cylinder of the compound engine?

A. Equal to that of the single condensing cylinder.

Q. Then, if the power could be got from one cylinder with the steam expanded a certain number of times, would it pay to make the more expensive compound engine with two cylinders to expand the same number of times?

A. No.

Q. Where then has the compound condensing engine any advantage over the single cylinder condensing engine?



JOY VALVE GEAR. SWING LINK DESIGN.

A. The steam at high temperature and pressure entering the high pressure cylinder of a compound engine is not chilled by the low temperature of the cylinder due to the low terminal pressure; also the thrusts of the compound cylinders on the crank pins and crank shaft are more uniformly distributed around the axis than where there is but one cylinder.

Q. Who invented the principle of the compound engine, and when?

A. Jonathan Hornblower patented it in 1781, but could not apply it because Watt's patents were in existence.

Q. Who first applied the principle of the compound engine, and when?

A. Arthur Woolf patented in 1804 the earliest compound engine in which the principle was practically carried out.

Q. In Woolf's early engines, how were the cylinders placed?

A. Both cylinders were at one end of a working beam; the condensing cylinder at the outer end, and the high pressure cylinder closer up, with less stroke.

Q. What was the principal objection to this Woolf engine?

A. All the power was applied on one side of the centre of the beam.

Q. How and when was this fault first remedied, and by whom?

A. By Wm. McNaught, in 1845, putting the condensing cylinder at one end of the beam and the high pressure cylinder between the main centre and the crank.

Q. What is another good arrangement of a Woolf engine?

A. By making the cylinders horizontal and side by side, secured to a single bed plate; both piston rods having one cross-head in common, one connecting rod taking the whole power to the crank.

The first column = P = initial pressure of the steam above a perfect vacuum on entering small cylinder; the second = R = $\sqrt{\frac{P}{t}}$ shows the relative areas of the two cylinders, and also the number of expansions in high-pressure cylinder; the third column, = P = the terminal pressure in high-pressure cylinder, gives the pressure at beginning of stroke in condensing cylinder; the fourth column, = H, contains the hyperbolic logarithms of R; the fifth, = S, gives the

A TABLE SHOWING THE RELATIVE AREAS OF THE TWO CYLINDERS OF A COMPOUND ENGINE, WITH THE AVERAGE PRESSURE IN EACH CYLINDER, ETC.

P	R	P'	H	S	p'	p
30	2.449	12.25	.896	23.23	7.52	15.71
35	2.645	13.22	.972	26.10	7.83	18.27
40	2.828	14.14	1.040	28.85	8.04	20.81
45	3.000	15.00	1.098	31.47	8.22	23.25
50	3.162	15.86	1.150	34.00	8.40	25.60
55	3.316	16.58	1.197	36.44	8.58	27.86
60	3.464	17.32	1.242	38.83	8.74	30.09
65	3.605	18.02	1.281	41.13	8.85	32.28
70	3.741	18.70	1.319	43.39	9.08	34.31
75	3.872	19.36	1.353	45.57	9.11	36.46
80	4.000	20.00	1.386	47.72	9.24	38.48
85	4.123	20.61	1.415	49.78	9.34	40.44
90	4.242	21.21	1.444	51.85	9.44	42.41
95	4.358	21.80	1.470	53.84	9.56	44.28
100	4.472	22.36	1.497	55.84	9.64	46.20
105	4.582	22.91	1.521	57.77	9.73	48.04
110	4.690	23.45	1.545	59.69	9.82	49.87
115	4.795	23.98	1.567	61.57	9.89	51.68
120	4.898	24.45	1.589	63.43	9.96	53.47
125	5.000	25.00	1.609	65.32	10.05	55.27
130	5.099	25.50	1.629	67.02	10.13	56.89
135	5.196	26.00	1.647	68.77	10.21	58.66
140	5.291	26.46	1.665	70.51	10.26	60.25
145	5.385	26.93	1.683	72.25	10.32	61.93
150	5.477	27.38	1.700	73.95	10.38	63.57
155	5.567	27.84	1.716	75.67	10.46	65.21
160	5.656	28.32	1.732	77.28	10.52	66.76
165	5.744	28.72	1.748	78.93	10.58	68.35
170	5.830	29.15	1.763	80.56	10.64	69.92
175	5.916	29.58	1.777	82.14	10.70	71.44
180	6.000	30.00	1.791	83.73	10 75	72.98
185	6.082	30.41	1.805	85.32	10.80	74.52
190	6.164	30.82	1.818	86.86	10.85	76.91

average pressure during a stroke in a single cylinder, for the different values of R and $= P\frac{1+H}{R}$; the sixth column, $=p'$, gives the average pressure during a stroke in the condensing cylinder of a compound engine, and $=P'\frac{H}{R-1}$ and the last column, p, gives the average pressure during a stroke in the high-pressure cylinder $P=\frac{1+H}{R}-P'\frac{H}{R-1}$.

Now, as the power to be given out by both cylinders is to be the same, the power that is required to be given out by $A=\frac{95}{2}=47.5$ H. P., and as $\sqrt{\frac{60}{5}}=3.464$, $t \times 3.464=17.32=P'$, and from the above formula we get $p'=17.32\frac{H}{R-1}=8.74$ lbs. average pressure per square inch on piston. So that we can now get what area of piston is required to give out this power by

$$\frac{47.5 \times 33.000}{400 \times 8.74} = 450 = 24'' \text{ diameter;}$$

and as the area of the two pistons are to each other as 1 to 3.464, we get the area of small piston = 130 square inches = 13″ diameter.

Q. How many principal forms of compound engine are there?

A. Two; one where the exhaust from the first cylinder goes direct into the second (Woolf type), and one where there is an intermediate receiver.

Q. How many kinds of piston movements are possible in receiver compound engines?

A. Three—

1. Entire independence.

2. Equal number of strokes per minute, with initial points in certain relation, and

3. Further limited by periods of tarrying (as in the Worthington pump).

Q. How do the pistons move in the Woolf type?

A. Together. The steam from the top of the first cylinder exhausts into the bottom of the second, and *vice versâ*.

Q. How do the pistons of a receiver engine move?

A. In a receiver engine, the pistons are connected to cranks on one shaft, at right angles to each other.

Q. Comparing the Woolf and the receiver types of compound engine, which is the more economical of steam?

A. When there is no clearance and no intermediate fall of pressure, there is no difference in the steam economy of the Woolf and the receiver types, but when there is an intermediate fall of pressure, the work done on the receiver system is greater than by the Woolf type.

Q. In a compound engine, how is the actual rate of expansion in the first cylinder, found?

A. To find the actual rate of expansion in the first cylinder, divide the period of admission plus the clearance, into the length of stroke plus the clearance. Thus, suppose cut-off at $\frac{1}{4}$ and clearances at one end equal to $\frac{1}{10}$ the piston displacement, then the actual expansion rate would be, (just as in an ordinary non-compound engine,)

$$1.05 \div 0.25 = 4.2.$$

As a formula, $R' = \frac{L'}{l'}$

Q. How do you find the ratio of intermediate expansion in a compound engine (receiver type)?

A. The ratio of intermediate expansion may be got from the pressure before and after expansion. Thus, if the fall of pressure was from 20 lbs. (*absolute*) in the first cylinder, to 15 lbs. in the second, the ratio would be $20 \div 15 = 1.333$.

Q. Does the volume of the receiver give any evidence as to the expansion between the first and the second cylinders?

A. No.

Q. How may we determine the rate of ntermediate expansion between first and second cylinders, in a Woolf engine?

A. Either by the method necessary in the case of a receiver engine, or from the volume of the intermediate space.

Q. How do you find the ratio of expansion in the second cylinder in a Woolf engine?

A. Find the ratio of the capacity of the first cylinder, plus its clearance, plus the

intermediate space, to the capacity of the second cylinder plus the intermediate space, (taking this last to include the clearance of the second cylinder).

Q. Is the ratio of expansion in the second cylinder of a receiver engine affected by clearance?

A. No.

Q. How do you find the actual expansion rate in the second cylinder of a receiver engine?

A. Where there is no intermediate fall of pressure, the actual expansion rate in the second cylinder is the ratio of the cylinder volumes. Where there is an intermediate fall of pressure, the ratio of actual expansion in the second cylinder is reduced. Thus, if the cylinder ratio is 3, and the ratio of the intermediate fall to the final pressure in the first cylinder is 4, then the actual expansion rate in the second cylinder will be $(4-1) \times 3 \div 4 = 2.25$.

Q. How do you find the total actual expansion rate for a Woolf engine?

A. Multiply the ratio of the first to the second cylinder by the length of stroke, and divide by this length plus the clear-

ance; add to the quotient the ratio-value of the intermediate space; multiply this sum by the actual ratio of expansion in the first cylinder. Thus, suppose cut-off in the first cylinder at $\frac{1}{4}$; clearance $\frac{1}{20}$; cylinder ratio 4; ratio value of the intermediate space $\frac{1}{3}$. Then the actual expansion rate in the first cylinder will be $1.05 \div 0.25 = 4.2$. The modified rate of the cylinders is $4 \times \frac{1}{1.05} = 3.809$; and $3.809 + .333 = 4.142$. Finally, $4.2 \times 4.142 = 17.396$ = total actual ratio of expansion.

Q. How do you find the combined actual ratio of expansion behind the pistons, in the Woolf engine?

A. Multiply the actual ratio of expansion in the first cylinder by the ratio of expansion in the second cylinder. Or divide the product representing the total actual ratio of expansion by the ratio-value of the intermediate space, plus one. Thus, if the total actual ratio of expansion is 17.396, and the ratio-value of the intermediate space is $\frac{1}{3}$, then $17.396 \div 1.333 = 13.05$, the combined actual ratio of expansion behind the pistons (in a Woolf engine.)

Q. How do you find the total actual expansion-ratio in a receiver engine?

A. Multiply the ratio of the first and second cylinder volumes by the actual expansion rate in the first cylinder. This gives the total actual expansion rate in a receiver engine.

Q. How do you find the combined actual ratios of expansion behind the pistons of a receiver engine?

A. Multiply the actual expansion rate in the first cylinder by the expansion rate in second cylinder. That is, multiply the ratio of the first and second cylinder volumes by the actual ratio of expansion in the first cylinder and by one less than the ratio of the intermediate fall of pressure to the final pressure in the first cylinder.

Q. How do you find the net work done in the two cylinders of Woolf engines, for one stroke, with a given combined actual ratio of expansion?

A. Add one to the hyperbolic logarithm* of the given combined actual ratio of expansion; multiply the sum by the period of admission to the first cylinder, in feet, plus the clearance, expressed in feet, of stroke; from the product subtract the clearance;

* See table of hyperbolic logarithms, p. 141 of Part I.

multiply the remainder by the net piston area of the first piston in square inches, and by the initial cylinder pressure in lbs. per square inch. This gives the net work in foot lbs. *per stroke.*

Thus :—suppose the net area of the first piston, 500 square inches ; of the second, 1,500; initial pressure, 63 lbs. (*absolute*); cut off at 2′ of the stroke, of 6′ clearance volume at one end, equal to 7%.

The actual ratio of expansion in the first cylinder is 6.42÷2.42=2.653. The ratio of intermediate expansion we assume as 1.333. The actual expansion rate in the second cylinder is equal to the ratio of the capacity of the first cylinder plus its clearance plus the intermediate space, to the capacity of the second cylinder plus the intermediate space; this last including the clearance of the second cylinder ; or

$$\left(3 \times \frac{6}{6.42}\right) + .333 \div 1.333 = 2.353.$$

The modified ratio of the cylinders is $\times 5 \div 6.42 = 2.804$; and $2.804 + .333 = 3.137$, which multiplied by $2.653 = 8.322$, the total actual ratio of expansion. The combined

ratio of expansion behind the pistons is 8.322÷1.333=6.242, the hyperbolic logarithm of which, as found in the proper table, is 1.8310.

Then the net work in foot lbs., during one single stroke, is (1+1.8310)=2.8310; 2.8310×2.42=6.851; 6.851—0.42=6.431; 6.431×63×500=202,576.5 foot lbs.

As a formula:—

$$w=aP[l'(1+\text{hyp. log. } R'')-c]$$

Q. How do you calculate the net work done in the two cylinders of a receiver engine for one stroke, with a given actual ratio of expansion in the first cylinder?

A. Multiply the first cylinder's actual expansion rate by the cylinder ratio; and to the compound ratio of the two cylinders add one; multiply the sum by the initial period of admission into the first cylinder, plus the clearance in feet. Call this product (*A*). Divide one less than the ratio of the two cylinders by the actual expansion rate in the first cylinder. Add one to the quotient, and multiply the sum by the initial clearance in feet. Call this product (*B*). Subtract product *B* from product *A*. Call the remainder (*C*). Multiply the net

piston area of first cylinder in square inches by the total initial pressure in lbs. per square inch, *above vacuum*, and by the remainder *C*. The product is the net work in foot lbs. for one single stroke.

Thus :—Suppose net area of first piston 500 square inches; of second, 1,500; total initial pressure above vacuum, 63 lbs. per square inch; clearance=0.42′; cut-off at 2′; stroke 6′; then actual expansion rate in first cylinder=6.42÷2.42=2.653; second expansion ratio 3; compound ratio 2.653×3=7.959, the hyperbolic logarithm of which is 2.0743 (as found in table;) 3.0743×2.42=7.44=*A*; (3—1)÷2.653=.7162; 1.7162×.42=.737=*B*; 7.44—.737=6.703=*C*; 63×500×6.703=211,145 foot lbs., the net work during one single stroke.

Q. What is the difference between a Woolf and a Wolff compound engine?

A. The Woolf compound engine has no receiver; the Wolff has a receiver.

DRYNESS OF STEAM.

Q. Is saturated steam at the condensing point or at the generating point?

A. Saturated steam standing over water,

s at both the condensing point and the generating point.

Q. When does steam approach to the condition of a perfect gas?

A. When superheated.*

Q. What are the advantages of superheating steam?

A. The advantages of superheating are (1) To increase the efficiency of the steam without producing a dangerous pressure; (2) To "improve the vacuum," that is, to lessen the density required to overcome a given resistance, and lower the back pressure; (3) To prevent condensation during expansion.

Q. How far is superheating carried with ease?

A. To 100° above the normal steam temperature due to the pressure.

Q. What can be said about the money loss by radiation from cylinders?

A. Radiation of heat from uncovered or unjacketed cylinders walks off with profits, and most inexcusably; for there are few makes of engines which cannot be cheaply covered with a non-conducting lag-

* See pages 87 to 90 of Part I.

ging, in such a way as that all parts can be readily disconnected and got at. Felt and other lagging materials are plenty in the market; and air-space, which is very desirable between even the best lagging and the cylinders, costs nothing.

Q. Are steam traps any use in giving drier steam in the cylinder?

A. Steam traps appear to some to be a needless expenditure. But we notice that any one who has had a cylinder head knocked out by reason of water in the cylinder generally believes in dry steam, and in steam traps, ever thereafter.

Q. What advantage has dry steam over wet, as regards friction?

A. Dry steam gives less friction in steam and exhaust pipes and passages, than wet; hence gives higher initial pressure and less back pressure.

Q. How much "primage" may there be in steam?

A. Hirn's experiments show an average of five per cent; Zeuner states it as seven and a half to fifteen per cent.

Q. What is "commercially dry" steam?

A. Steam containing only three to fou

per cent of water cannot be distinguished, by observation, from that which is perfectly dry, and may therefore be said to be "commercially dry."

THE GOVERNOR.

Q. Which requires the more sensitive governor, a throttling engine, or an automatic; and why?

A. The automatic, because the pressure in the cylinder varies more during the stroke than in a throttling engine.

Q. Which needs the more sensitive governor, an early cut-off, or a late cut-off; and why?

A. The early cut-off, because of the greater variation in pressure during a stroke.

Q. What is the objection to the ordinary centrifugal speed-governor?

A. The trouble with the ordinary centrifugal speed-governor is that it "has to go slow in order to go fast." This is true of any speed-governor; it does not operate to check the speed until the engine has actually speeded up beyond the regulation rate. As a consequence, it takes a turn or two of running at abnormal rate, before the

engine can be brought down to regular speed. Now, suppose an engine running 120 turns and belted on to a "dynamo" making 1,200 turns; the engine may make one turn at the rate of 150 turns per minute, and then be choked down to one turn at the rate of 90, and then be speeded up to one at 140, then at 100, then at 130, then 110, and then settle down to 120, as if nothing had happened. A speed indicator applied during ten seconds would indicate an average speed of 120 turns, and the engine would be supposed to be a marvel of steady running; but the dynamo would have been running at rates from 1,500 down to 900, if the belt slip did not prevent some of these variations.

Q. How might the objection to the ordinary centrifugal speed-governor be remedied?

A. What is wanted is a governor which will govern by the load and not by the speed, and will govern so quickly that if the engine is speeded at 120, *each turn* shall be made in one-half second; no spurts, no lags, no dancing.

This can be accomplished in the case of

an engine driving a dynamo, by "shunting off" a part of the current, through a coil passing around a vertical soft iron bar attached to the throttle valve or cut-off lever. Any attempted increase of speed would increase the current in the coil, and raise the bar in the coil, thus actuating the throttle or the cut-off, in the veriest fraction of a second.

Q. What classes of work require the best governors?

A. Electric light machines, textile machinery (particularly for silk), flour mills, and saw mills.

Q. What law governs the revolution of the ordinary rotating conical pendulum governor?

A. One revolution is performed in the same time as two vibrations of a common oscillating pendulum, having a length equal to the height of the point of suspension of the conical pendulum above the plane in which the balls revolve.

Q. How long an oscillating pendulum will make a vibration in one second?

A. 39.1393″ in the latitude of London.

Q. How long an oscillating pendulum

would vibrate in $\frac{1}{70}$th of a minute in the latitude of London?

A. $\left(\frac{\sqrt{39.193} \times 60}{70}\right)^2 = 28.75''$.

Q. How do you determine the number of revolutions that a conical pendulum governor will make, when the balls are at a given distance below the suspension point?

A. Multiply the square root of the height of the cone in inches by 0.31986, and this gives the time of revolution in seconds.

Q. How do you determine the diameter of the circle described by the balls of a conical pendulum?

A. Divide 187.58 by the number of revolutions per minute, and the square of the quotient will be the distance in inches, of the plane of revolution, below the point of suspension. Take the square of this vertical distance from the square of the length of arm in inches, and the square root of the remainder is the radius of the circle in which the centres of the balls revolve.

Q. Has the weight of the governor balls any influence upon the governor speed?

A. No.

Q. What may be said about the governor speed?

A. The governor should in no case make a number of revolutions per minute, exactly a multiple of the rotation speed of the engine, as that causes "dancing," and "plays hob" in mills where regular speed is a necessity.

Q. In guaranteeing the regulation of a steam engine, what should be specified?

A. In guaranteeing the regulation of an engine, the extremes and the averages of variation of pressure, speed and load, should be specified.

For instance, an engine may be rated to run within 2% of 120 revolutions, provided the governor is never set to keep the speed below 110 or above 135; and the pressure is kept between 75 and 85 lbs., averaging 78; under a load averaging 65 H. P., and never running below 55 nor over 75 H. P.

DUTY OR ECONOMY OF AN ENGINE.

Q. What is properly meant by the "duty" of an engine?

A. The amount of work done, in relation to the steam consumed; though it is

generally understood as meaning the amount of work done, in relation to the fuel consumed by the boiler supplying the steam.

Q. What is the only means by which steam can do useful mechanical work?

A. By changing the volume of the vessel containing it.

Q. How is the efficiency of steam in a perfect engine reckoned?

A. By dividing the range of temperature worked through, by the maximum initial absolute temperature of the fluid entering the cylinders.

Q. How is the efficiency of an engine as a machine reckoned?

A. By the ratio of the quantity of work transmitted from the engine to the machinery of transmission, to the work done by the steam in the piston.

Q. How should the economy of an engine be expressed, and why?

A. The economy of an engine should be expressed in lbs. of dry steam per hour per H. P., because expressing economy in lbs. of fuel per hour per H. P. depends on the boiler too, rather than on the engine

alone. Thus one boiler might require 600 lbs. of coal to evaporate 3,600 lbs. of water from 62° to 310° F., and produce say 120 H. P. in a certain engine; another boiler might require only 335 lbs. of coal to produce the same amount of steam and power.

Q. In buying an engine, when the rating is given in lbs. of steam per hour, what should be specified?

A. In rating an engine. in lbs. of steam per hour, it should be specified that the steam should be either saturated, dry or superheated*; and the initial pressure, point of cut-off, and load should be stated.

Q. Under what conditions does insufficient expansion cause waste?

A. Insufficient expansion is an evil which, especially in ordinary slide-valve engines cutting off at ¼ stroke and later, is responsible for a great deal of the waste. The higher the terminal pressure, as compared with the mean effective pressure,* the greater the loss from this source.

Q. How far should expansion be carried?

* See under these heads in Steam Engine Catechism, Part I.

A. In what is conventionally known as a "perfect" engine (in which the waste is *only* about two-thirds the entire amount of heat), expansion should be carried on until, in the case of a condensing engine, the terminal pressure would be as low as that corresponding to the temperature of the condensers. In the average non-condensing engine, we may say practically that the best economy of steam is obtained by expanding down until the terminal pressure is just as low as the back pressure.‡ Thus, roughly, if the back pressure is 2 lbs. above the atmosphere=16.7 lbs. absolute,§ and the initial pressure¶ in the cylinder was 68.8 lbs. by the gauge=83.5 lbs. absolute, the actual expansion rate‖ should be at least 83.5÷16.7=5; that is, not reckoning clearance, cut-off at least as early as one-fifth stroke. The steam user may readily prevent and cure this source of waste.

Q. Does expansion pay when there is a good vacuum?

A. The better the vacuum the greater the gain by expansion.

Q. Does expansion pay when there is a great deal of clearance?

A. The greater the clearance, the less the proportionate gain by cut-off at a fixed point of the stroke; because clearance practically lessens the expansion ratio.

Q. How far should expansion be carried?

A. In what is conventionally known as a "perfect" engine (in which the waste is *only* about two-thirds the entire amount of heat), expansion should be carried on until, in the case of a condensing engine, the terminal pressure would be as low as that corresponding to the temperature of the condensers. In the average non-condensing engine, we may say practically that the best economy of steam is obtained by expanding down until the terminal pressure is just as low as the back pressure. Thus, roughly, if the back pressure is 2 lbs. above the atmosphere=16.7 lbs. absolute, and the initial pressure in the cylinder was 68.8 lbs. by the gauge=83.5 lbs. absolute, the actual expansion rate should be at least 83.5÷16.7=5; that is, not reckoning clearance, cut-off at least as early as one-fifth stroke. The steam user may readily prevent and cure waste from a wrong grade of expansion.

Q. How much gain is there in adding lap so as to cut-off at half stroke?

A. If we suppose a theoretically perfect condensing engine, having no clearance, throttling, wire-drawing, *cylinder* condensation, nor steam or exhaust lead, and a perfect vacuum, we would get a theoretical

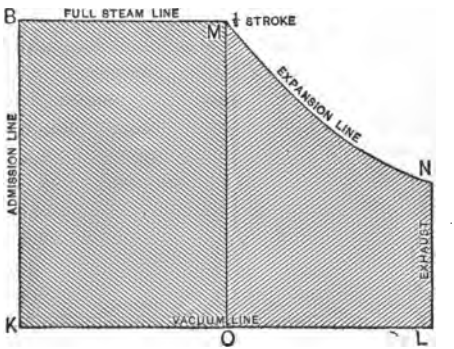


Fig. 6.

card like Fig. 6. when cut-off is at half stroke, and instantaneous. There would be an instantaneous rise of pressure, from perfect vacuum at K, to full boiler pressure at B; then the boiler pressure line would run at a uniform height, B M, until cut-off took

place (at half stroke in this case). At that point the pressure above vacuum would fall at such a rate that at stroke end it would be only half what it had been between commencement of stroke and point of cut-off. Then it would at once fall to vacuum, and remain at that until the end of the return stroke.

Such a diagram would be represented by the figure K B M N L; the height of the figure above the line K L representing pressures. The pressure at commencement of stroke is represented by B K; that at half-stroke, where cut-off takes place, by M O; that at stroke end, L N. The areas of the figures K M B O and O M N L represent the amount of work done before and after expansion, respectively. In this case it will be found that O M N L is 0.69 (or nearly seven-tenths) the area of K B M O; and that is the amount gained by cutting off at half stroke, *under the conditions named.*

Suppose that there was not a perfect vacuum, then instead of the pressure during the return stroke being nothing, it would be an appreciable quantity; and the

amount of work done would, instead of being K B M N L, be only A B M N D—as shaded. Of this, the part A B M R represents the work done during full steam, and the part R M N D, that done during expansion. (See Fig. 7.)

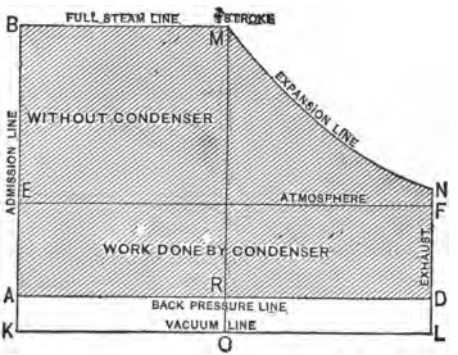


Fig. 7

In this case, the pressures above vacuum are the same as in Fig. 6, but the gain during expansion is not proportionately so great.

The better the vacuum, the greater the proportionate gain by expansion.

Now, suppose that this last case was altered by the introduction of clearance, say 2½%, (as shown in Fig. 8).

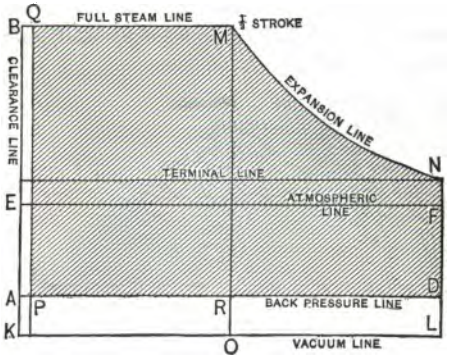


Fig. 8.

If Q M or P R is half of the whole stroke P D, then B M O K will be more than half of the whole volume, and N L will be more than half the initial pressure, K B. We have plainly marked all the lines of this diagram. It will be seen that the work done during expansion, (represented by the area R M N D) is not quite as large a proportion of the work done during full

steam, represented by the area Q M R P, as if there had been no clearance.

The greater the clearance, the less the proportionate gain by cut off at a fixed point of the stroke; because clearance practically lessens the expansion ratio.

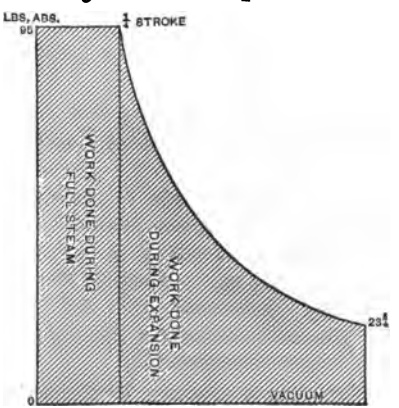


Fig. 9.

I will supplement my answer to the above question by showing what the proportions would be between the work done during full steam and that done during expansion, supposing all the conditions

to be as in Fig. 6, except that cut-off is at one-fourth stroke instead of at one-half. In this case there is much more work done during expansion than before (see Fig. 9).

The earlier the cut-off, under the conditions named, the greater the economy of steam by expansion.

But economy of steam is not necessarily economy of money.

Q. Is it not generally admitted that ¼ stroke is the most economical point of cut-off?

A. That depends upon the conditions. For instance, with initial pressure 27 lbs. by the gauge and cut-off at ¼, in a non-condensing engine, expansion would go below the atmospheric line, and during about the last third of the stroke the piston would drag on the crank pin.

Q. How much economy is there in high initial pressures?

A. I clip from my note book the following table showing the decreased steam consumption, consequent on increased initial pressure at such a point of cut-off as to give the stated mean effective pressures:

INITIAL PRESSURE.	M. E. P.	Lbs. Water per Hour per H. P.	Gain of Power per increment of 10 Lbs.	Economy of Fuel per increment of 10 Lbs.
10 Gauge Lbs.....	9.3	75.3	per ct.	per ct.
20 "	16.3	42.9	76	43
30 "	21.2	33.	30	23
40 "	25.6	27.3	20	17
50 "	29.1	24.	14	12
60 "	32.	21.9	10	9
70 "	34.5	20.3	8	7
80 "	36.3	19.2	6	5
90 "	38.1	18.4	5	4
100 "	39.2	17.8	3	3

Q. Is there any plainly evident proof of the economy of high pressure steam?

A. That high steam pressures tend to economy is plainly shown in ocean steamers. Twenty-five years ago they carried 16 lbs. pressure per square inch, and it took 5 to 6 lbs. of coal per hour to make a H. P. Now they carry 75 lbs. and get one H. P. per hour with 2½ to 3 lbs. of coal. The "Anthracite," carrying 350 to 500 lbs. pressure, crossed the Atlantic with only 1 lb. of coal per hour per H. P.

Q. Under what circumstances may small ports be more economical of steam than large ones?

A. In some cases (as, for instance, in locomotive engines with badly designed valves and valve gear) where small ports, by wire drawing the steam, practically effect the expansion of the steam before the lap of the valve cuts off the admission.

Q. How is it that some people find inside lap to save coal and others find it do just the opposite?

A. The effect of inside lap, or exhaust lap, as it is variously termed, is to diminish the exhaust passage, and after a certain portion of the stroke to completely close it. Thus we have two different effects upon economy.

The first is a tendency to produce back pressure if the engine be running very fast; and back pressure previous to compression is always a waste of steam. So that if one valve had too little travel, or too little arch in the valve, or too small an exhaust passage, cutting out inside lap might free the exhaust and thus increase the economy of steam.

The second effect of exhaust lap is to cushion the exhaust at a certain portion of the return stroke; and this cushion or compression, if in proper amount, has for its effect to save the waste steam contained in the clearance spaces, and which would otherwise reduce the grade of expansion and also chill down the steam entering from the steam-chest. The higher the speed (so long as the exhaust passages are large enough and there is enough exhaust lead) the greater the benefit of exhaust lap simply as giving a cushion to the reciprocating parts.

The greater the waste clearance space, the greater the benefit of exhaust lap up to that point, at which point the exhaust steam in the waste spaces is compressed up to steam-chest pressure. The higher the steam-chest pressure the greater the benefit of exhaust lap, because the greater the compression of exhaust steam in the waste clearance space.

Thus both may be right.

"Of two evils, the least is always to be chosen."

Q. What are the causes and effects of wire-drawing?

A. Unsteady wire-drawing has as its cause, insufficient size of ports and passages, and as its injurious effect, reduction of the mean effective pressure and of the ratio between this and the terminal pressure.

Q. Does wire-drawing necessarily wholly represent wasted energy?

A. Wire-drawing does not entirely waste energy, as while it lowers the initial pressure it causes slight super-heating; but it lessens the ratio between terminal and back pressures, and lessens the expansion ratio.

Q. What is the most advantageous adjustment of compression or cushion?

A. The most advantageous adjustment of compression or cushion, (from the point of view of steam economy, and entirely irrespective of the question of pounding) is when the quantity of steam cushioned just fills clearance spaces at initial pressure.

Q. Why will not compression save all the loss by clearance?

A. Compression cannot save all the loss by clearance, because the space through which the piston travels during compression is lost for the production of work, while friction continues.

Q. What is the amount of compression to give, for maximum steam economy?

A. For every point of cut-off and for every amount of back pressure, there is with each per cent of clearance, a point of compression which gives the best steam economy.

The following table by D. K. Clark, gives the best period of compression with 7% clearance, and for several back exhaust pressures and for several points of cut-off:

COMPRESSION OF STEAM IN THE CYLINDER; BEST PERIODS OF COMPRESSION.

Total back pressure, in parts of the total initial pressure at vacuum.	Cut off in percentages of stroke.						
	10	20	30	40	50	60	75
	Period of compression, in parts of stroke.						
Per cent.	per ct	per ct	per ct	per ct	per ct	per ct	per ct
2½	65	52	42½	35	29½	23½	16
5	57	47	39	32½	27	22½	15
10	45	40	34	29½	25	20½	14
15	37	33	29	25½	22½	18½	13½
20	—	27	26	23	21	17	13
25	—	23	22	21	18½	16	12½
30	—	—	19½	19	17½	15½	12
35	—	—	17½	16½	16	14½	11½

Q. What is the advantage of exhaust pre-release?

A. Exhaust pre-release enables the entire fall of pressure to take place towards the end of the stroke, effecting a saving in one part of the stroke and a loss in another. By a proper point of release the saving may be made to considerably exceed the loss.

Q. What is the point for exhaust release, at which the greatest saving is effected?

A. The greatest saving in work is effected by making the release at such a point that one-half the fall of pressure shall take place at the end of the forward stroke and one-half at the commencement of the return stroke.

Q. What is the effect of engine room temperature on cylinder condensation?

A. The lower the engine room temperature, the greater the amount of cylinder condensation.

Q. What are the advantages of a small engine over an underloaded large engine?

A. As the friction of the engine and transmission, does not greatly vary with the load; as the friction of a small engine is

less than that of a large one driving the same load, it is best to use a small engine at maximum economic capacity, than a large one, underloaded.

The following figures are given by Mr. Julius L. Hornig as from indicator cards:

COMPARISON TABLE OF USEFUL PERCENTAGE UP TO FULL CAPACITY:

Work divisions.	1	2	3	4	5	6	7	8	9	10
Per cent for large engine.	35	55	66	75	80	86	92	95	98	100
Per cent for small engine.	55	75	86	95	100					
Per ct. difference.	20	20	20	20	20					

Thus the small engine, with only half the capacity of the large one, shows a gain of 20% if used instead of the large engine, when the latter is worked at half its capacity or less.

Q. How does the exhaust waste fuel?

A. Every steam engineer should look upon the exhaust as one of the most active and competent of the seven thieves which are robbing his engine of heat and his

pocket of cash. The exhaust waste is by reason of the heat taken up by the exhaust from the interior surfaces of the cylinder walls, which thus, instead of remaining hot, become chilled and cause internal condensation during admission and expansion.

Q. Is lagging the exhaust pipe any use?

A. No.

Q. Is air space between lagging and cylinder, advisable?

A. Yes.

Q. Does the loss of a steam engine by friction vary with the load?

A. Not necessarily.

Q. What is the effect of a draft through the engine room?

A. To increase cylinder condensation.

Q. Which is the more economical method of lowering the capacity of an engine:—throttling, or cutting off earlier?

A. In general, earlier cut-off is the more economical of steam, because throttling causes wire-drawing and lowers initial pressure, while earlier cut-off only diminishes the mean effective pressure.

CAPACITY OR POWER.

Q. What is meant by the "capacity" or "power" of an engine?

A. The capacity or power of an engine is the amount of work it can do, irrespective of economy. Thus, a 22″ × 36″ engine running 110 revolutions (660′ piston speed) per minute, has, at 80 lbs. initial pressure, a capacity of about 397 H. P.; but it is more economical, that is, uses less steam per hour per H. P., when worked to from 225 to 342 H. P.

Q. How may the horse power of a steam engine be expressed?

A. The horse power of a steam engine may be expressed by Prof. Marks' formula:

$$(HP) = \frac{PLAN}{33{,}000};$$

(HP) being the indicated horse power; P, the mean steam pressure on piston, in lbs. per square inch; L, the stroke in feet; A, the piston area in square inches, and N, the number of strokes (or double the number of crank revolutions) per minute. The

author has a somewhat shorter formula:

$$(HP)=\frac{PAT}{33,000};$$

in which the term T, representing the piston travel in feet per minute, replaces the two terms L and N, in the preceding formula.

Q. What is a nominal horse power?

A. "Nominal horse power" does not mean anything in particular. It corresponds to "as long as a piece of string;" or "as big as a lump of chalk."

Q. What is an actual or real horse power?

A. An actual or real horse power is 550′ lbs. per second, or 33,000 foot lbs. per minute, or 1,980,000 foot lbs. per hour.

Q. What is a "*force de cheval*" or "*cheval vapeur*?"

A. The "*force de cheval*" or "*cheval vapeur*," is the French horse power. It is

		ft. lbs.
75 kilogrammeters	per second	=542½
or 4,500 "	' minute	=32,549
270,000 "	" hour	=1,952,932

or about $\frac{1}{70}$ part less than the British horse power.

Q. How can you ascertain the gross horse power from the piston area, quickly?

A. Look in the following table, in the square corresponding to the mean effective pressure and piston speed of your engine, and you have a number, which, multiplied by the piston area, gives the gross horse power.

MULTIPLIERS FOR VARIOUS SPEEDS AND PRESSURES:

Lbs.	Lineal piston speeds in feet per minute.				
M. E. P.	300	400	500	600	700
10	.090909	.121212	.151515	.181818	.212121
15	.136364	.181818	.241818	.272727	.318182
20	.181818	.242424	.303030	.363636	.424242
25	.227273	.303030	.378788	.454545	.530303
30	.272727	.363636	.454545	.545455	.636364
35	.318182	.424242	.530303	.636364	.742424
40	.363636	.484848	.606061	.727273	.848485
45	.409091	.545455	.681818	.828283	.954545
50	.454545	.606061	.757576	.909091	1.060606
55	.500000	.666667	.833333	1.000000	1.166667
60	.55. 455	.727273	.909091	1.090909	1.272727

Thus:—an engine 10″ bore, with 40 lbs. mean effective pressure and 400′ piston speed, has a gross horse power of 78.54× .484848=37.75.

Q. What is the "factor of horse power" of an engine?

A. The factor of horse power of an engine is the product of the area of the piston in square inches, by its speed in feet per minute, divided by 33,000. This figure, multiplied by the mean effective pressure, gives the gross horse power of the engine. Thus, a 14″ × 24″ engine, making 150 turns per minute, has a factor of horse power of

$$\frac{(14 \times 14 \times .7854) \times (2 \times 300)}{33{,}000} = 2.79$$

and with 40 lbs. mean effective pressure the gross horse power is $40 \times 2.79 = 111.7$.

I contributed the following some time ago to the *Millstone*, and reproduce it in order that my readers may construct similar tables for their own use:—

"We often see even engine builders take a long while ciphering up the horse power of their standard sizes of engines at various speeds and mean effective pressures, and wonder that they do not have a "factor of horse power" calculated for each engine at its rated rotatum speed, so that its horse power, with any given mean effective pressure, may be obtained in a moment by one simple multiplication. We give herewith

the factors of horse power of a series of engines that we get up for "direct driving," electric light machines, circular saw mills, &c. It will be noted that the rotation speeds are unusually light; but the engines are guaranteed to run cool and steady, and govern closely even under the excessive changes of load usual in such work.

The factor of horse power is from "Grimshaw's Formula" $\frac{P A T}{33,000}$; being $\frac{A T}{33,000}$; that is, the quotient of 33,000 into the piston area (in square inches) times the travel (in feet per minute).

Diameter.	Stroke, Inches.	Revolutions.	Piston Speed.	Area.	Factor Horse Power.	Horse Power at 50 lbs. Mean effective pressure.
3	3	1,200	600	7.07	.12855	6.427
4	4	900	600	12.57	.22855	11.427
5	5	750	625	19.64	.37196	18.598
6		750	625	28.27	.53571	26.785
6		650	650	28.27	.55683	27.841
8		525	700	50.27	1.06606	53.303

Q. What is the "horse power constant" of an engine?

A. The "horse power constant" of an engine is generally understood to mean the horse power developed for each revolution per minute and each pound per square inch mean effective pressure. Thus, a 12″×21″ "Hartford" engine, under an initial steam pressure of 80 gauge lbs., and cutting off at ¼ stroke, (say 35 lbs. mean effective pressure) would, at 170 revolutions per minute=595′ piston speed per minute, have a horse power constant of .0187; and its horse power would be 170×35×.0187= 111.27.

Q. What is an easy rule for gross and net horse power of engines of 12″ bore?"

A. *In all engines of* 12″ *bore* the gross horse power equals the products of the mean effective pressure and the travel in feet per minute, divided by 291.8; and the net horse power is about equal to the mean effective pressure times thrice the lineal piston speed, with three figures pointed off from the right.

Thus:—12″ engine with 40 lbs. mean effective pressure, and 400′ piston speed;

then the net horse power=40×1,200=48.

A simple formula for horse power of engines is

$$\frac{PAT}{33{,}000}$$

P being the mean effective pressure in pounds per square inch, A the piston area in square inches, and T the piston travel in feet per minute.

Q. Is there any usual mean effective pressure, and piston speed, which makes it especially easy to calculate gross horse power from piston area?

A. Referring to the table,* it will be seen that the multiplier for 55 lbs. mean effective pressure, and 600′ piston speed, is 1; so that in any engine working at those rates (which are customary inactual practice) the gross horse power is the same as the square inches of piston area.

Q. Is it fair to rate the power of an engine by its ability to drive a certain machine, rated to take a given horse power?

* This table will be found on page 87.

A. No. The indicator, dynamometer, and the brake offer the only fair method of rating an engine. For instance, some descriptions of electric light machines sold to produce a given candle power with a stated number of horse power, consume two to ten times the stipulated motive force.

Q. What is a hyperbolic logarithm, and how can I find the logarithm or hyperbolic logarithm of any number?

A. You understand that a "power" of a number is that number multiplied by itself a certain number of times. Thus $2\times2=4$ is the *second* power of 2; $2\times2\times2=8$ is the *third* power of 2; $2\times2\times2\times2\times2=32$ is the *fifth* power of 2. Sometimes they are written thus:—2^2 for second power of 2; 2^3 for third power of 2; 2^5 for fifth power of 2, and so on. The little figure showing how many times the number is taken as a factor is called the "exponent," or "shower."

Now note that if you multiply 2^2 or the *second* power of 2, by 2^3 or the *third* power of 2, you get 2^5 or the *fifth* power of 2; that is, in multiplying powers of a

number together you *add* the exponents.

Conversely, in dividing one power of a number by another power of a number you *subtract* the exponent of the divisor from the exponent of the dividend. Thus, 2^7 divided by 2^3 gives 2^4; that is, $2\times2\times2\times2\times2\times2\times2$ (or 128) divided by $2\times2\times2$ (or 8) gives $2\times2\times2\times2$ (or 16).

The "reverse" of a power is a "root." That is, if 25 is the second power of 5, then 5 is the second root of 25; if 81 is the fourth power of 3, then 3 is the fourth root of 81.

A root may be expressed by writing its exponent as the denominator of a fraction up at the right hand top corner of the number; thus $81^{\frac{1}{4}}$ means the fourth root, or reverse of the fourth power, of 81.

Now suppose we want to take the third power of the fourth root of 3; or the fourth root of the third power of 3. We can express that $3^{\frac{3}{4}}$; that is a fractional exponent. Let us work out the fourth root of the third power of 3, so we can see how much it is: $3^3=3\times3\times3=27$.

$3^{\frac{3}{4}}=27^{\frac{1}{4}}$, got by the following operation:

27.00′00(5.1961 is the *second* root of 27.
25.

101)200
101

1029)9900
9261

10386)63900
62316

103921)158400
103921

5.19′61)2.2794 is the *second* root of 5.1961 or the fourth root of 27, or the fourth root of the third power of 3, or is $3^{\frac{3}{4}}$, or is $\sqrt[4]{3^3}$, as it may also be written.
4.

42)119
84

447)3561
3129

4549)43200
40941

45584)225900
182336

Now suppose we write $\frac{3}{4}$ decimally as 0.75, and put it $3^{0.75}$; then we have a decimal exponent for the 3.

You can easily see that every number must be some root of some power of some other number; that is, must be an exponent (generally a fractional exponent) of some other number taken as a "base."

Thus, if we take 10 as a "base," then 2 is equal to the hundred thousandth root of the 30,103d power of 10, and may be written $10.^{0.30103}$; 3 is equal to $10.^{0.47712}$; and 6, or 3×2, equals $10.^{0.77815}$. If you will add 30,103 and 47,712 you will get 77,815.

Now there are tables prepared which show what power and root of 10, every number is; so that if we want to multiply any long number by any other long number, we turn to the table, get these "exponents," add them, and then look in the table and see what number has for its exponent of 10, the sum of these exponents, and that number is the product of the two numbers.

These exponents of 10 are called common logarithms; there are plenty of tables of

them in mathematical works, and they save an immense deal of labor, besides ensuring greater accuracy than where long calculations are gone through with.

Here we "do an example" in multiplication by common logarithms:

Multiply 1.23456, of which the com. log. is.... 0.090967
By 7.89383 " " " " " " 0.897563

The sum of the logarithms is..... 0.987634
of which the "natural number" is 9.919177.

Now we'll try an example of *division* by logarithms:

Divide 187.8562, of which the com log. is..... 2.272041
By 34.5999, of which the com. log. is..... 1.537944

The difference of the logarithms is..... 0.734097
of which the natural number is 5.421225.

Now to take any power of a number: Suppose we want the fifth power of

184.9263, of which the common logarithm is.. 2.266999
Then we multiply that logarithm by..... 5

The product is.....11.334995

having for its "natural number" 216,269,203,980.09+, which is the fifth power of 184.9263, as you can prove by covering a few square yards of paper with figures.

To take any root of a number: Suppose we want the seventh root of 9,999.999, of

which the common logarithm is 3.999999. Dividing 3.999999 by 7 we get 0.571428, which is the common logarithm of 3.7276. Then 3.7276 is the seventh root of 9,999.999.

The "hyperbolic logarithm," or "Napierian logarithm," of any number is the "common logarithm" multiplied by 2.3025851; there are tables of hyperbolic logarithms in most of the important works which treat of steam, and they are just as essential to the steam engineer as the common logarithms are to the scientific man who works in other lines. They *shorten labor.*

Thus, suppose an actual expansion rate of two in a cylinder 2′ long, and one square foot area of piston. Then at point of cut off we have one cubic foot of steam. Say it is at 100 lbs. pressure above vacuum, and the expansion is perfect. Then at full stroke we have 2 cubic feet of steam at 50 lbs. At ¼ stroke we have 1.5 cubic feet at 66⅔ lbs. At ⅓ stroke we have 1.25 cubic feet at 80 lbs. At ⅕ we have 1.75 cubic feet at 57.14 lbs.; and so on.

To *roughly approximate* the average pressure in the cylinder during the time of expansion, we may thus proceed:

At $\frac{1}{8}$ stroke it is.....80 lbs.
" $\frac{2}{8}$ " " .....66.67 "
" $\frac{3}{8}$ " " .....57.14 "
" $\frac{4}{8}$ or full stroke it is.....50. "

4)253.81 lbs.

Rough approximate average....63.45 lbs.

But suppose we count it at the $\frac{1}{16}$ths instead of at the $\frac{1}{8}$ths, it ciphers up

At $\frac{2}{16}$ stroke there are $\frac{2}{8}$=1.125 cubic feet; pressure $\frac{8}{9}$ of 100= 88.88lb
At $\frac{4}{16}$ stroke there are $\frac{4}{4}$=1.25 cubic feet; pressure $\frac{4}{5}$ of 100= 80. lb
At $\frac{11}{16}$ stroke there are $\frac{11}{8}$=1.375 cubic feet; pressure $\frac{8}{11}$ of 100= 72.73lb
At $\frac{8}{16}$ stroke there are $\frac{8}{5}$=1.5 cubic feet; pressure $\frac{8}{12}$ of 100= 66.67lb
At $\frac{13}{16}$ stroke there are $\frac{13}{8}$=1.625 cubic feet; pressure $\frac{8}{13}$ of 100= 61.54lb
At $\frac{14}{16}$ stroke there are $\frac{14}{8}$=1.75 cubic feet; pressure $\frac{4}{7}$ of 100= 57.14lb
At $\frac{15}{16}$ stroke there are $\frac{15}{8}$=1.875 cubic feet; pressure $\frac{8}{15}$ of 100= 53.33lb
At $\frac{16}{16}$ or full stroke there are $\frac{16}{8}$=2 cubic feet; pressure $\frac{8}{16}$ of 100= 50. lb

8)530.29lb

Second approximate average, 66.28lb

Now the table of hyperbolic logarithms says that the hyperbolic logarithm of 2

(the expansion rate) is .6931472; so that without all the foregoing figuring, which after all only gives a rough guess at the result, we may get the exact average pressure as 69.31472 lbs.

THE CONDENSER

Q. What is a vacuum?

A. A vacuum is an absolutely empty space; although as we ordinarily understand it, it means a closed space from which a part of the contents has been abstracted, and which contains nothing but air or other gaseous fluid (or a mixture thereof), at a less tension than that of the atmosphere.

Q. What is the pressure in a vacuum?

A. The pressure or tension in a perfect vacuum is nothing, outwards; but there is an unbalanced pressure of nearly 15 lbs. per square inch (at the sea level) from without, towards its center, tending to fill it with whatever fluid surrounds it.

Q. What is it which exerts this unbalanced pressure of nearly 15 lbs. per square inch from without inwards, on the vessel containing a vacuum?

A. The weight of the atmosphere. A

column of air of one square inch at the base, and running from the sea-level to the upper limits of the atmosphere, weighs 14.7 lbs.

Q. With what velocity does air rush into a perfect vacuum ?

A. 1,338 feet per second.

Q. What is the use of the so called air pump, in a condensing engine ?

A. The "air pump" removes the water from the condenser to prevent the air which it contains, accumulating and still further destroying the vacuum.

Q. Could a condensing engine be worked with a steam pressure less than that of the atmosphere ?

A. Yes, if once started.

Q. What would be the objection against running a condensing engine with steam at less than 14.7 lbs. per square inch in the boiler ?

A. The boiler could not be blown out; and the gauge cocks could not be tried.

Q. When and by whom was the condenser first applied to decrease exhaust pressure ¿

A. By James Watt in 1765.

Q. How many principal types of condensers are there; and what are they?

A. Four:—the surface condenser of marine vessels:* the air pump jet condenser;† the ordinary siphon condenser, and the exhaust steam induction condenser.

Q. What is a "siphon condenser?" ‡

A. A "siphon condenser" is one in which the exhaust steam escapes through a nozzle of special shape, surrounded by another through which flows with a fall of 34′ an annular current of water which condenses the exhaust steam.

Q. What is an exhaust steam induction condenser?

A. A modification of the siphon condenser in which is utilized the velocity of the exhaust steam. It provides for its own water supply without the aid of a pump.

Q. How is the exhaust steam induction condenser started?

A. By a small jet of live steam, which is shut off after the machine is started.

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* See Steam Engine Catechism, page 74.

† See Steam Engine Catechism, page 73.

‡ See Steam Engine Catechism, page 78.

Q. What is essential in a so called siphon condenser?

A. That the condensing water shall properly fill the cone or jet, but without overcrowding. If the cone be too large, too much water will be needed.

STEAM ENGINE RATING.

Q. What are the principal elements to be considered in choosing or comparing steam engines?

A. The principal elements to be considered in choosing or comparing steam engines, are their duty or economy, simplicity, regulation, and compactness.

Q. What are the relative values of these four principal elements?

A. This varies with the use to which the engine is to be put, and the conditions under which it is to be run. Opinions of engineers and engine buyers differ widely. The following opinions from eminent authorities will prove of interest in this connection:

The Harlan & Hollingsworth Co., Wilmington, Del., builders of marine engines, while saying that the conditions indicated by such figures are necessarily modified in

different engines, estimate as follows: Economy, 40; durability, 20; simplicity, 20; regulation, 25; and compactness, 5.

Mr. J. W. See, Hamilton, Ohio, does not think it proper at all to divide a possible 100 points among the features stated on any scale, as some other features entitled to recognition in the hundred might prove to have been omitted; and he doubts any one built to give each feature its proper percentage. He would say that a percentage based upon a possible hundred of each feature now thought of or turning up in the future would be nearer right. In six out of every ten engines Mr. See thinks that regulation would be a matter of no import whatever, and in some economy would be nothing.

The Norwalk Iron Works, South Norwalk, Conn., by E. Hill, Treasurer, say that regarding the rating of engines they would judge that no fixed rule could be adopted. In their opinion for engines intended for use in large cities and vicinity, economy would stand first, and in crowded buildings compactness would always be an important feature. Simplicity there is of not much

account, especially on a large engine, as skilled help is easily obtained. Durability would stand low in the scale, as repairs are easily made, and proprietors are progressive and ready to introduce improved machines as they appear.

For flour mill engines and many similar industries, regulation is the first point to be considered, as the quality of the entire product depends upon that. The other points would then be affected by other circumstances of the case. For a coal mine durability stands pre-eminent, and next, simplicity, and if the power is transferred to a long distance through wire ropes, regulation should be first-class.

Mine engines on barren mountain peaks, difficult of access, demand durability and economy.

The Atlas Engine Works, Indianapolis, Ind., say concerning the value of the various features that make up the performance of an engine, they suppose that they would vary with different classes of engineers for different services. For stationary engines for ordinary land purposes their ideas would be to divide the 100 points thus:

Economy, 40; durability, 25; simplicity, 10; regulation, 20; compactness, 5.

Prof. de Volson Wood says, in this connection, that special conditions will greatly affect in particular elements. For average cases he thinks that adjustability will be of but little value, and in the hands of an ignorant runner it may be a damage and should (for him) be marked 0; a few intelligent runners might prize it highly. Similarly, in some cases, a want of compactness would render the engine nearly worthless for that particular case.

Such being the case, the judgment of different engineers, Prof. Wood thinks, will differ greatly, but if he were to rate the elements with a definite number, instead of giving a range, he would mark them about thus: Economy, 30; durability, 40; simplicity, 15; regulation, 5; compactness, 10. Still, he is not certain how much may be included in regulation in this schedule—such as automatic cut off, or mechanical means for taking up wear.

Messrs. O. A. Pray & Co., Minneapolis, Minn., rate economy at 50; durability, 35; simplicity, 5; regulation, 5; compaatness, 5.

The Fitchburg Steam Engine Company would divide about as follows: Economy, 50%; durability, 16; regulation, 16; simplicity, 13; compactness, 5. They think the question has many side issues upon which each depends; hence no fixed rule can be adopted for all conditions. Economy is dependent to a great degree upon close regulation under all variations of load and pressure, avoiding spasmodic use of steam; and also upon simplicity, for although, under critical tests, any engine may control well and all parts be in perfect adjustment, yet the lack of simplicity allows speedy and perhaps disastrous misadjustment, so that the difference between the critical and the average performance for five years may be largely in favor of a simple engine with a much lower first duty.

Mr. F. F. Hemenway says that in the little thought he has as yet been able to give the subject, he must confess that the difficulties, mainly, have presented themselves to him. For instance, economy with coal at $10, and with sawdust to throw away, would be two different things; in like manner regulation may be very important

in one instance and of not much in another. Mr. Hemenway says that if he were to do the same thing he would, if he wanted to make it generally acceptable to engineers, make his own figures to the best of his knowledge and then ask the opinions of all he thought best, requesting them to alter the figures and give reason for so doing.

DESIGN, CONSTRUCTION AND ERECTION.

[See also both in other parts of this Supplement, and in the Steam Engine Catechism, under the heads of the various parts of the engine.]

Q. What is a rotative engine?

A. One in which the reciprocation of the piston in a straight line causes the crank and shaft to turn.

Q. What is a rotatory engine?

A. A rotatory engine is one in which there is no reciprocating (or "to and fro") piston, but in which the action of the steam causes the shaft to turn, without the intervention of intermediate mechanism.

Q. What type of valve causes the most wire-drawing?

A. The slide valve causes the most wire-drawing.

Q. What are the advantages of the inverted vertical type of stationary engine?

A. Lower frictional resistance, smaller amount of floor space, higher speed, and less foundation needed, than for a horizontal engine. Besides, it can be erected in a more contracted position difficult of access.

Q. What is meant by a "Corliss" type of engine?

A. In the United States, an engine with the rotative Corliss valve; in Europe, any engine with cut-off regulated by the governor.

Q. What is the advantage of "conical" or tapered valves?

A. "Conical" valves have the advantage that they may have wear taken up by end adjustment.

Q. What is the objection to conical valves?

A. "Conical" (or tapered) valves, unless hung on centres or on hardened trunnions, wear faster at the large end, and leak there, while they cannot be "set up" without pinching the small end.

Q. What are the disadvantages of two-ported engines?

A. One disadvantage of having but one port for each admission and exhaust is that the cool expanded steam cools the metal of the port and lowers the temperature of the next admission.

Q. How are cylinders and steam-chests generally protected from external radiation?

A. By wooden "lagging" (made of strips held on by metal bands or covers) or by coarse hair felt with metal casing.

Q. Is cork a good lagging for steam pipes?

A. Cork is a good lagging for steam pipes if kept from charring by the interposition of asbestos paper or asbestos blanket.

Q. What is the disadvantage of having the eccentric keyed on the shaft?

A. Adjustment is very difficult.

Q. What is the disadvantage of having the eccentric fastened by set screws?

A. They are liable to slip.

Q. How is the amount of lead determined?

A. The proper amount of lead must be

determined by experiment with each engine. Twin engines run at differing speed rates or under differing pressures or with steam of differing dryness, require differing leads.

Q. What is "setting" slide valves?

A. Fastening the eccentrics in proper position on the shaft (or axle) and adjusting the length of eccentric rods and valve stems.

Q. Where should steam lead be greatest, and why?

A. Steam lead should be greatest on high speed engines and on beam engines, because these are most likely to thump, by reason of their greater momentum.

Q. Where should exhaust lead be greatest?

A. Exhaust lead should be greatest with high speed engines and late cut-off.

Q. Which is generally greater, exhaust lead or steam lead?

A. Exhaust lead is generally greater than steam lead, because exhaust lead has to release all the steam in the cylinder, while steam lead is only to fill the clearance with live steam.

Q. How should a slide valve be ground?

A. In grinding a slide valve in which there are hollow places, it is better to use water with the emery or sand than oil; and kerosene is better than any other oil for this purpose.

Q. What is necessary before either release or admission can take place?

A. The valve movement must exceed the exhaust lap before the exhaust is open; and must exceed the steam lap before admission can commence.

Q. How may the port opening for any amount of valve movement, be calculated?

A. To get the port opening for any amount of valve-throw, subtract from that throw, the steam lap or the exhaust lap, whichever governs the case.

Q. What should be the rule for determining steam port area?

A. The steam port area of an engine must be sufficient to give the steam ample inlet without wire-drawing, and exit without causing excessive pressure. The steam passages must not be so large as to cause excessive clearance. For a piston speed of 600′ per minute, the area of each port

ought to be about $\frac{1}{15}$ the piston area, with ordinary steam; $\frac{1}{13}$ will answer where the steam is very dry. For other piston speeds the area should be proportional.

Q. What easy, approximate rule can you give for area of ports of "square" engines?

A. For "square" engines running 600′ per minute, with 55 lbs. mean effective pressure, two square inches of port area per horse power, or $\frac{1}{100}$ square inch for every cubic inch of cylinder capacity, is about right; and other speeds and mean effective pressures, proportionally.

Q. What is the objection to large port area?

A. The objection to large port area is that it causes excessive clearance.

Q. What is the objection to small port area?

A. The objection to small port area is that it causes excessive friction.

Q. How is the steam port width determined?

A. The steam port width is found by dividing the necessary area by the greatest practicable length.

Q. What are the objections to a very narrow exhaust port?

A. A very narrow exhaust port will choke the exhaust at the end of the valve stroke.

Q. What are the objections to a very wide exhaust?

A. A very wide exhaust necessitates an unnecessarily long slide valve, and excessive friction.

Q. What is gained by having the steam-chest under the cylinder?

A. The special use of having the steam-chest under the cylinder, is to free the cylinder readily of water of condensation, which would otherwise endanger the cylinder heads.

Q, What are the evils of too small crank pin diameter?

A. Too slender a crank pin is liable to bend or break.

Q. What are the objections to too large a crank pin?

A. Too large a crank pin heats; one of the greatest troubles about a steam engine.

Q. What objections to a very long crank pin?

A. A very long crank pin is liable to bend.

Q. What is the disadvantage of a very short crank pin?

A. A very short crank pin has not enough bearing surface.

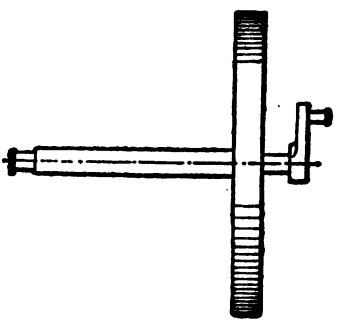

Fig. 9.

Q. What is the best proportion between fly-wheel weight and the work on the piston?

A. Mr. N. J. Raffard has examined into the best proportions between fly-wheel weight and the work on the piston; also the best place for the fly-wheel on the main shaft, as regards friction; and the superiority, from this point of view, of the horizontal over the vertical engine.

Supposing an engine carrying full steam, and with connecting rod of infinite length, and without fly-wheel, and such equal work as would be put on it by a completely universal screw propeller. For each position of the crank, the work carried by the connecting rod being constant in amount and direction, the pressure of the shaft against the brasses would be constant, and the friction

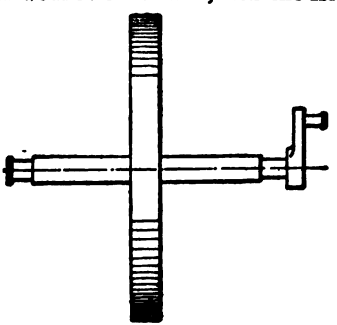

Fig. 10.

the same throughout the whole revolution, no matter what the inclination of the cylinder.

Adding a fly-wheel, as in Fig. 9, near the crank, its weight comes into play. In the vertical engine this fly-wheel weight helps

or hinders the piston, according to the direction of motion; but this helping or hindering, in the case of the vertical engine, balance. Supposing the same engine arranged horizontally; there will be 40% more fly-wheel friction.

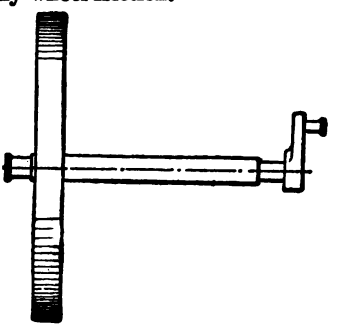

Fig. 11.

Placing the fly-wheel in the middle of the shaft as in Fig. 10, then in the vertical machine only fifty per cent. of the friction due to the fly-wheel weight will be neutralized; but in the horizontal engine, only 30%.

Putting the fly-wheel at the other end of the shaft (Fig. 11) the fly-wheel friction will be twice as much in the vertical engine as

in the same engine with fly-wheel next the crank.

Mr. Raffard applied these principles to the Woolf compound engine of the following dimensions, &c. :

Stroke,.....0.90 metres=35.433 in.
Diam. L. P. cyl.....0.40 " 15.748 "
" H. P. " .....0.18 " 7.087 "
" Shaft.....0.155 " 6.103 "
" Flywheel.....4.50 " 13.12 ft.

Con. Rod, 5 cranks long.

Pressure, 5 atmospheres=say 73.5 lbs.

Cut-off at $\frac{1}{4}$ stroke.

Rev. per min., 40 to 45.

Theoretical ind. H. P., 60 H. P.

Weight of flywheel, (see table.)

The table gives in kilogrammeters* the friction in the bearings, coming from the combined efforts of the pressure on the connecting rod, and the weight of the fly-wheel, under various conditions of fly-wheel position relative to the crank; and neglecting that of the shaft itself; though taking into account the obliquity of the connecting rod:

*1 kilogrammeter equals .672 lb. per ft. or 2.016 per yard.

Fly wheel position on shaft.	No. 1, Vertical Engine,		No. 2, Horizontal Engine.	
	Fly wheel weighing 2400 kilog† — Friction in kilog§	Fly wheel weighing 4800 kilog‡ — Friction in kilog	Fly wheel weighing 2400 kilog — Friction in kilog	Fly wheel weighing 4800 kilog — Friction in kilog
By the crank....	105.6	140.	117.5	162.2
Middle of shaft.....	138.4	172.2	139.6	184.2
Opposite crank.....	172.5	239.2	172.5	239.2

This table shows how much the friction can vary according to the disposition of the parts of the machine; and that the variation can run as high as 99 kilogrammeters per turn in an engine of about 30 H. P., even when the stroke is long as compared with the cylinder diameter.

This variation is considerably increased with the work in the pistons; that is to say, when with the same nominal power, the

† 4,891 lbs. av. ‡ 9,782 lbs. av.

§ Kilogram equals 2.2046 lbs. avoirdupois; 1 lb. avoirdupois equals 0.45359 kilograms.

stroke is diminished and the fly-wheel weight increased.

The Woolf "pendulum" engine with fly-

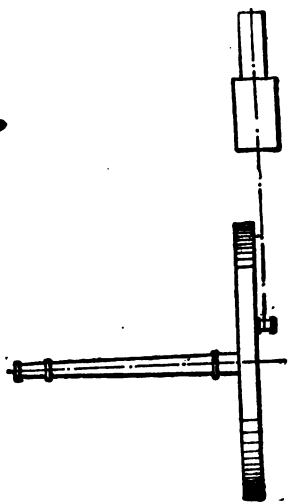

Fig. 12.

wheel near the connecting rods (see Figs. 12, 13, 14,) and vertical cylinders, permits of reducing or even completely neutralizing fly-weeel friction.

Q. What is the best place for the fly-wheel on the main shaft, as regarcs friction?

A. Next the crank. See account of

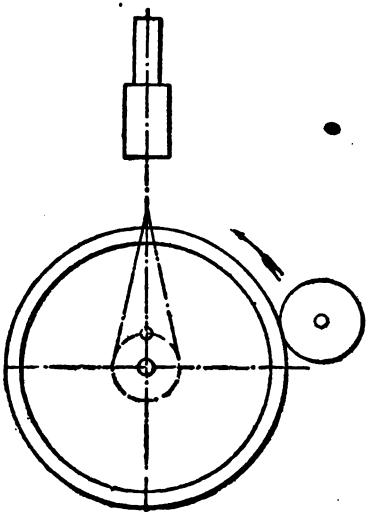

Fig 13.

Raffard's experiments, page 114.

Q. Which has the least fly-wheel friction, a horizontal or the vertical engine, other things being equal?

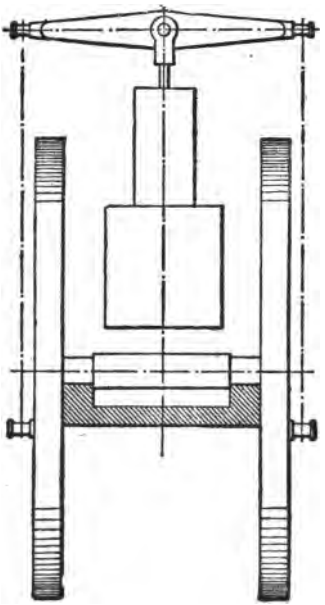

Fig. 14.

A. The vertical. See account of Raffard's experiments, page 114.

CARE AND USE.

Q. When and how should the valves be finally set?

A. The valves should be finally set by reference to indicator diagrams, and while they are hot.

Q. What is the best way to keep piston valves tight?

A. The best way is to bore the valve chamber accurately, make a good plug fit, and when leaks commence, rebore and make a new valve.

Q. How can slide valves be cured of wearing curved?

A. Slive valves that wear curved can generally be prevented from that sort of thing, after valve and seat have been planed up, by using a "back rod" as a guide?

Q, How may cutting of cylinders and excessive friction be lessened?

A. Dry plumbago will improve the running and decrease the cutting of steam engine cylinders.

Q. How may wire drawing be lessened?

A. Wire drawing may be lessened by the use of double eccentrics and "gridiron"

slide valves, and by increasing the area of steam ports and passages.

Q. How may wire drawing be avoided?

A. Wire drawing may be avoided by using the Corliss types of valve gear, where the admission valves are suddenly opened and closed.

Q. What may be said of the practice of having the parts of an engine "about true enough" rather than absolutely true?

A. The only place for a piston rod is in the exact centre of the piston and of the cylinder; the only position of the crank shaft is absolutely at right angles with the cylinder axis, and the crank pin should be absolutely in line with the shaft.

Q. How should the guides be lined up?

A. Guides on steam engines should be lined up by the bore of the cylinder and not by the counter bore.

Q. Which wears an engine out most, for a given piston speed, frequent reciprocation or fast rotation?

A. An engine wears out by the reciprocation of the parts rather than by the rotation of the wheel and shaft.

Q. What is "lost motion?"

A. Lost motion is looseness of parts which are supposed to prevent fixed relative positions. Thus, if the connecting rod have lost motion at the cross head pin the piston starts and "takes up" that looseness before it moves the connecting rod.

Q. What often causes jarring at the time of cut off?

A. Having an overhead supply pipe with a right angle in its horizontal portions.

Q. How may the pound in an engine be located ?

A. The pound in an engine may be located by placing one end of a piece of $\frac{1}{4}''$ wire, 8″ long, in the teeth and applying the other to each end of the crank shaft bearings, cylinder, etc. Where there is most shock there will be the pound.

Q. What is the probable cause of pounding when the crank is on the dead centre?

A. Pounding at the end of the stroke when the crank is on the dead centre may come from a ridge in the cylinder, or at the guide bar end, or from the connecting rod brasses not being keyed up tight enough.

Q. What is the probable cause of pounding at half throw?

A. Pounding when the crank is at half throw will not be caused by any of the above, but will be apt to be occasioned by the crank pin not being parallel to the crank shaft, or from one or the other of the crank shaft journals being low.

Q. Where do horizontal piston rods wear most?

A. Horizontal rods wear most in the middle; the ends wearing elliptical ("oval" so called); the back end wearing most on the bottom and the crank end most on the top (where the engine throws under this is reversed.)

Q. What are the requisites of a piston rod packing?

A. A piston rod packing must allow the rod to move freely up, down or sidewise with little friction, with little wear of the rod or of the packing; and must make a steam tight joint under the highest pressure that there will be found in the cylinder.

Q. What defects are we likely to find in piston rods that are out of true?

A. Pistons are apt to be out of centre, to

be out of round and crooked; either one or the other, and sometimes all three; in addition to which they are apt to be smallest in the middle, and the back end will be worn most upon the bottom and the crank end most upon the top.

Q. What are the principal causes of piston rod packing and valve stem packing being in bad condition?

A. Among the principal causes of rod packings being in bad condition are excessive speed, and steam pressure; lack of alignment; crooked or bad rod; too short stuffing boxes; packing of inferior kind or quality, or badly put in, or insufficiently cared for; and in the case of the piston rod packing, leaky piston or too little clearance.

Q. What are the effects of high pressure on many rod packings?

A. High pressure, generally accompanied by high speed, and always by high temperature, causes baking and abrasion of many packings, especially fibrous vegetable ones.

Q. What does lack of alignment do to rod packings?

A. Lack of alignment puts excess of pressure on one side of the packing and tends to wear it out rapidly at that side.

Q. How do crooked rods affect packings?

A. Crooked rods put sudden pressures first on one side and then on another, of the packings, and unless they are adjustable, tend to tear them out.

Q. What precaution should be taken in the use of fibrous piston rod packing?

A. If fibrous material be used, care should be taken lest it contain grit, which will scratch the piston lengthwise and cause channels for the escape of steam, besides making more friction.

Q. What rule is there for thickness of coil packing?

A. Coil packing should be in thickness, half the difference between the diameters of rod and box. Thus a 3″ rod in a 4½″ box requires ¼″ packing.

Q. What is the effect of cutting the rod packing rings too long?

A. If the rod packing rings are too long they will not hug the rod, and the packing will not be steam tight.

Q. What is caused by cutting the rod packing rings too short?

A. If the rod packing rings are cut too short they will not meet, and leakage will result.

Q. What is the proper procedure in packing a stuffing box?

A. In packing a stuffing box with coil packing, put in as much packing as will just allow the gland to enter; screw up hard; then slacken.

Q. What should be done if the eccentric straps heat?

A. If the eccentric straps heat, the bolts should be slackened a very little; and when the engine stops the straps should be removed and scraped (not filed) to a true and smooth surface.

Q. What is the best cure and prevention of heated bearings?

A. Good dry "air floated" plumbago (black lead; graphite) *free from grit*, will cool any hot bearing that anything else will, and will prevent heating, better than anything else which we have ever seen.

Q. How can bright work best be protected?

A. Bright work may be much better protected by tallow and lime than by white lead and tallow. In any case there should be an excess of tallow, so that the covering may be readily wiped off with waste. I have used the German *putz-pomade* for brass and nickel plate, with great satisfaction.

Q. How can bright work be freed from rust?

A. Crocus cloth properly used, with a soft pine stick, removes rust from bright work before it has time to get deep into the material, and should be kept in every engine room. It must be remembered that *rust breeds rust.*

Q. How should tight nuts be moved?

A. By first loosening them with coal oil, if rusted.

Q. How often should indicator cards be taken?

A. Indicator cards should be taken daily from every engine of twenty-five H. P., or over. They *pay* in time, money, freedom from annoyance, &c.

Q. What important memoranda may be made concerning the care and use of the indicator?

A. Test your indicator by some standard gauge before taking any cards. Never connect the indicator to the blow-off hole in the cylinder. Indicate both ends of the cylinder, and the steam chest as well.

Q. What is the advantage of a small steam chest?

A. A small steam chest gives the throttle better control of the engine than a large one, because if there is a large one the steam therein may act by expansion between the point of complete throttling and that at which the lap of the valve causes cut-off. A small steam chest has also to recommend it, that it decreases loss of temperature and pressure from condensation.

Q. What may be said of the carrying powers of steam pipes of different diameters?

A. Doubling the diameter gives *six* times the carrying power for long distances, as proved by the experience of the New York Steam Heating Co.

Q. What should be the diameter of the exhaust pipe?

A. The exhaust pipe should be double the area of the steam pipe.

Q. How may the exhaust be deadened?

A. The exhaust of an engine may be deadened by running it through a large box filled with perforated heads, or with ordinary pebbles all of one size. If the box is large there will be no back pressure made. Another way is to utilize the exhaust to heat the feed water.

Q. How are slide valves and seats finished?

A. By planing, then filing, and scraping so that every part of the valve face touches the seat, in all positions of the valve.

Q. What objection is there to cast cranks?

A. Cast cranks will not permit of shrinking the crank pin in the eye, without danger of cracking.

Q. Are half round brasses in connecting rod ends, good things?

A. Half round brasses in connecting rod ends must have been designed by old Clootie. Avoid their use.

Q. Which is the greater cause of crank pins heating; increase of speed or increase of pressure? and why?

A. Within certain limits, an increase in

speed heats the crank pin more than a proportionate increase in pressure, because it expels the lubricant more rapidly.

A. What is the best way to put in a crank pin?

A. The best way to put in a crank pin is to taper it a little and drive it in slowly by hydraulic pressure.

Q. How may the shock of a sudden admission of high pressure steam be neutralized?

A. By making the piston and cross head very heavy so that the shock will be absorbed by reason of the inertia of the then motionless or nearly motionless mass of metal.

Q. Is the velocity of the crank pin and fly wheel constant at all parts of the stroke?

A. It should be; and is, practically.

Q. Is the velocity of the piston the same at all parts of the stroke?

A. No.

Q. Is the variation of the velocity of the piston regular or irregular?

A. Regular.

Q. When is the velocity of the piston the least?

A. At stroke ends, when it is absolutely *nothing*.

Q. When is the velocity of the piston the greatest?

A. When it is a little ahead of mid-stoke on the forward stroke of a stationary engine, and a little behind mid-stroke on the return stroke.

Q. What is the maximum speed of the piston?

A. The maximum piston speed is about $1\frac{1}{7}$* of its mean speed.

MISCELLANEOUS.

Q. What are the advantages of steel piston rods?

A. Steel piston rods have to recommend them—greater smoothness than wrought iron rods and greater stiffness for a given diameter; or for a given stiffness—less weight and less reduction of the effective area of the piston on the crank side.

Q. How is the piston rod fastened to the piston head?

A. Its end is tapered and passes through

*Accurately $\frac{3.1416}{2} = 1.5708$.

a tapered hole in the piston head. The taper prevents its being pushed through too far, (even if there be no shoulder, as there generally is) and a key or a nut prevents its being pulled out.

Q. How is the piston head of a large engine generally constructed?

A. Of two cast iron (or bronze) pieces; one a "spider," to which the rod is fastened; the other a "follower plate" bolted to the spider by "follower bolts."

Q. How is the piston made to work steam tight in the cylinder bore?

A. By "packing" of one kind or another, in suitable circumferential grooves. Sometimes hemp strands are employed. Most usually there are cast iron split rings set breaking joint and spring into the grooves in the spider. These rings are kept out by either their own elasticity, or the steam pressure in the cylinder, or springs within the spider.

Q. How are the piston rod and the valve stem generally made to work steam tight in the cylinder head and valve chest end, respectively?

A. By passing them through a stuffing

box, consisting of a cylindrical chamber surrounding the rod or stem, so as to leave an annular space, which is filled with packing material, (vegetable or metal) and the pressure of which against the rod is adjusted by a "gland," fitting outside the rod and inside the stuffing box, the depth to which it penetrates, being regulated either by bolts or by the stuffing box and the gland being threaded.

Q. What should be required of rod packing?

A. Rod packing should accommodate itself without undue wear of rod or packing, to crooked, tapering, and fluted rods.

Q. How is the piston rod attached to the cross head?

A. By its end being tapered and inserted into a tapering hole in the cross head, and held in by a key.

Q. How is the connecting rod attached to the cross head?

A. To a pin, generally called a *wrist pin*, and cast in one piece with the cross head, each end bearing in one side of the "fork" of the cross head. (In the Straight Line engine the wrist pin is fast in the

connecting rod end and turns in bearings, at each end, in the cross-head fork.)

Q. Which guide wears the most, in a horizontal engine; and why?

A. In a horizontal engine which "runs over," the bottom guide wears the most, because on the out stroke the cross-head end of the connecting rod bears down, the connecting rod being in compression, and the crank pin above the center line; and on the return stroke the connecting rod is in tension and the crank pin below the center line, so that the cross-head end of the connecting rod is then pulled down.

If the engine "runs under" as in a locomotive engine when going ahead, the top guide gets the most wear.

Q. What may be said of the width of piston rings and piston-valve rings?

A. Piston rings and piston-valve rings should be wider than the ports which they pass, to prevent their catching in the latter.

Q. What are the ways of cutting piston packing rings?

A. The usual way is by a straight diagonal cut but this opens out as the cylin-

der bore enlarges. We prefer a cut in this shape:

Q. What principles should be carried out in regard to designing and making piston heads?

A. "The piston head should be deep enough to insure tightness, durability and good alignment, without being unduly heavy, especially in vertical engines. It is a good plan to make it of two portions cast from the same pattern, but one having turned on it a hub corresponding to a recess in the other. These two may be bored parallel, smaller than the rod, which is to be turned down parallel to fit the hole accurately, care being taken to leave no square corner at the junction of the full-sized and the reduced portions. It is cheaper to bore parallel than to turn the rod taper and ream a taper hole.

"The ring should be cast larger than the head; three water grooves turned in it, and a sufficient portion cut out diagonally to bring it down to the right diameter when

sprung in to the recess turned for it in the head."—*Power.*

LUBRICATION.

Q. How should lubricants be regarded?

A. Lubricants should be regarded as money savers, not as necessary evils. They should not be selected because of cheapness per gallon, but by reason of fitness for the special purpose.

Q. Will good lubrication save steam and fuel, or only ensure cooler running and greater durability of the engine?

A. I have for some years been trying to persuade the average lunkhead engine-owner and pig-headed "engineer" (courtesy title, in too many cases of which all know, for "starter and stopper") that there is no economy in first sloshing an engine all over, inside and out, with oil and then letting it run dry and cutting; that it takes more oil, more coal and more repair-bills, (to say nothing of time lost from shut downs) than where a decent lubricating device is used. But it is rarely that even with a good coal saving as a test I can prove the fact. 'Tis a fact all the same.

Q. What should be the characteristics of a good lubricant?

A. A lubricant should have body enough not to be squeezed out under great pressure; and yet be fluid enough to enter between tightly fitting bearing surfaces; should be a good conductor of heat, and have a high power of reducing friction. It should not be liable to decompose in the air or in use nor to corrode the bearings and journals, &c. It should vaporize at a high temperature and chill or thicken at a low; and of course be free from grit and adulteration.

Q. What are the best lubricants?

A. Sperm is the best general lubricant, but for a given friction-reducing power costs too much. It is generally better to pay half as much for a lubricant with two-thirds the anti-frictional and durable qualities that sperm has.

For light bearings the most fluid oil that will remain in place is the best.

Lubricants should adhere naturally to the bearing metal. In this respect mineral oils are first, then sperm, then neats-foot and then lard.

Q. What is the effect of heat on lubricants?

A. Bearings should never be allowed to *commence to heat*, as heating reduces the consistency of lubricants and causes them to squeeze out and cause still more heating, and so on.

Heating should be *prevented*, rather than cured. It cracks, distorts and abrades the surfaces in contact; and even sets fire to the lubricant; overheating, softening and weakening both the journal and the bearing, and even welding them together.

Q. Does it pay to use poor lubricants?

A. No. A saving of 10% in cost of lubricants has been shown by the indicator to cause an increase of 10% in the more important item of coal.

Q. What are the legitimate uses and proper modes of use of oil cans?

A. Oil cans have their legitimate uses and modes of use. They are not intended to squirt oil at an oil hole; but where it is necessary to oil through a small orifice the nose of the spout should be put in the hole, not simply near it. The "illuminating oil can" seemed to be one of the—the *very*

few—things which really "filled a long felt want;" but for some reason or other it has not got to general public knowledge, much less into general public use.

Q. How may hot bearings be cooled down?

A. Sulphur and grease have a cooling tendency upon hot bearings. This is probably because the fine metallic dust formed by the hot journal combines with the sulphur to form a greasy sulphide.

Q. How should the guides and slides be oiled?

A. From cups on the cross head and also from stationary cups on the slides.

Q. What is a good way to oil crank pins and their brasses?

A. By having the crank pins bored and drilled for oil passage and connected with an oil tube parallel with the crank and extending as far as the center of the main shaft, where it has a filling cap. The faster the crank turns, the greater the centrifugal force tending to drive the oil from the closed end at the shaft center, to the crank pin end.

Another good way is by a wiper on the

crank, which licks a drop of oil from a metal or other strip, fed from an oil cup, every time the crank goes over.

Sometimes there are "oil cellars" attached to the under side of the straps; these being metal boxes filled with oil.

Q. How may the crank pin of a portable engine be kept oiled?

A. A simple arrangement for oiling the crank pin of portable engines is to make it of a steel tube, and fill it with tallow. By this means it may be made a perfect self oiler.

Q. How should oil be supplied to the cylinder?

A. Oil should be supplied to steam cylinders continuously and in small quantities. Attention to this will result in saving lubricant and in keeping the cylinder in better condition.

Q. What can be said of cylinder oils containing mineral oils?

A. Cylinder oils containing mineral oil are apt to be entirely free from all honeycombing action on the valve seat and cylinder wall; but when the piston has insufficient bearing surface, and the pressure is

great, they sometimes have not enough "body" to prevent cutting.

Q. What can be said of cylinder oils for high pressures and high speeds?

A. Cylinder oils for high pressures and speeds should be better than those for low. This is seldom the case. Tell your oil dealer what you need and see that he gives it to you.

Q. How should cylinder oils be tested?

A. The only way to test a cylinder oil is by an indicator, in the cylinder of an engine (preferably the one on which it is desired to use it) under the speeds and initial pressures of everyday use. Run the engine with some standard oil, as lard, tallow, &c., take friction cards and load cards; then run for a day with the oil to be tested, and then commence taking cards, with the same load on as before.

[The following query was made in reference to the question in the Catechism, concerning light piston head *versus* light cylinder head:]—

Q. Isn't there a misprint on page—of the Catechism where you state that a *piston*

head should be made light, so that, if it brought up against something, it would break instead of the cylinder head? This is contrary to locomotive and marine engineering. Some locomotive builders turn a narrow groove between the joint and bolt holes. Here on the Mississippi the flange on the head is made from $\frac{1}{8}''$ to $\frac{1}{4}''$ less in thickness than flange on cylinder. I think smashing the piston wrecks the engine more than smashing the head, because you can make a wooden cylinder head do, but a wooden piston would not last long.

A. The light head was advisedly recommended; for engines having late cut off because it has less inertia, hence reverses more easily than a heavy head; it should be cheaper to replace than a cylinder head; and the risk of the cylinder itself is something worth considering. It is a common thing in locomotive and stationary engines to have a heavy piston not only smash the cylinder-head but tear out part of the flange and wall of the cylinder proper. The narrow groove between joint and bolt holes is probably for the purpose of laying in a copper wire, to make a tight metal joint, or letting

a rubber gasket fill in. The flange on the head should be thinner than that on the cylinder, to cause the head flange to break rather than the cylinder; and the piston head should be lighter than either. A wooden piston would run for a while as well as a wooden head; the greater the diameter, the more this is so: and the consequences of running a wooden piston head are much less severe than those of blowing out a wooden cylinder head; particularly on shipboard, where there is more risk of scalding the engineers to death and causing a panic among passengers. A wooden head for the crank end of the cylinder would be difficult to get up and supply with stuffing box.

The following communications on this subject appeared in POWER for September 1885.

MESSRS EDITORS:—In reference to the subject, (page 153, of POWER, for August) of light piston heads *vs.* weak cylinder heads for quickness and cheapness of repair, in case of smashing by their coming together or getting water, nuts, etc., between them;—the question seems to be raised whether the cheapest of the two

parts named should not be made the weakest, and enough so to be certain to break in case of any of the accidents alluded to, and to break easily enough and completely enough to save the piston from all damage.

Unquestionably, it would be preferable that only one of the parts should be damaged rather than both, and also that the part broken should be the cheaper one. But unfortunately, the part most difficult to save by such means, namely the piston, is the most expensive one. If only water was to be provided for, it would be a matter of no difficulty to make the parts of such relative strength that the head would yield, but when a nut gets in it falls to the bottom and catches the piston where it is weakest and the head where it is the strongest, and if the latter were made so weak that the rod would not even be bent it would not be strong enough to properly resist the working pressure, still less those occasional whacks on a little water that impose considerably more strain on the parts than that due to the steam pressure, yet far from enough to do any damage to parts made reasonably strong. But since

water is by far the most frequent mischief maker, it may be asked, why not provide for it? To this it may be answered that unless the piston was made very massive and strong the degree of weakness in the head necessary to insure its breakage would be such that the legitimate stress from steam pressure with the help of the frequent light water concussions that all engines are liable to, would be sufficient to gradually "fatigue" the metal of the heads, that is, to gradually weaken it, so that it would, in time, give out from ordinary service. To avoid this "fatigue" is the function of the "factor of safety," well known to constructing engineers, which is generally placed at about five times the estimated breaking strength.

Still it may be said that given a piston with a factor of safety of, say ten, the head might have the required factor of safety of five and still be certain to break easily enough to save the piston, which is probably true. The difficulty in the way of obtaining the proper strength is that it would be almost impossible for the designer to determine it theoretically on paper, with

sufficient accuracy. Every different size and design of engine would have to have several of its heads broken by hydraulic pressure, enough, in fact, to ascertain not only the proper strength for a given quality of iron, but also the margin of uncertainty, arising from the unavoidable variation in the strength of the iron used.

A far better plan for such engines as contain distributing valves that will not yield to dangerous water pressures is to provide them with spring valves that will yield to a pressure slightly in excess of the maximum working pressure. Then neither piston nor cylinder head will be broken.

J. W. THOMPSON,
with Buckeye Engine Co.
Salem, Ohio.

Mr. Wm. Lee Church, M. E., formerly with the Buckeye Engine Co., and now of the firm of Westinghouse, Church, Kerr & Co., (Westinghouse and Reynolds-Corliss engines) says:—

"Personally I am of the opinion that less damage results in horizontal double acting engines from breaking the piston than from breaking the cylinder head. The piston is

apt to break up into small pieces and the subsequent movements of the rod under the action of the fly wheel will not usually do any damage. If the piston is made so strong as to wreck the cylinder head it is sometimes apt to crack the cylinder itself under the flanges. This will almost assuredly take place if the breakage occurs when the piston is moving towards the shaft. With the Buckeye engine we used to habitually make the piston the breaking piece; with a single acting engine, however, like the Westinghouse, all danger of breakage is necessarily on the up stroke, and we therefore make our pistons very strong and weaken the head artificially by a pop-out plug."

Mr. Wm. A. Harris takes the opposite view of the case; as may be seen by the following:—

MESSRS. EDITORS:—In my judgment, it is preferable to have a weak cylinder head in preference to the piston head, as in the latter case, if the piston was wrecked it would cut the cylinder, and very likely destroy it, while if the cylinder head was the weakest

and would give way, you would save your cylinder.

WILLIAM A. HARRIS.

Providence, R. I.

Q. Can any table be got up, showing the proportional average effective pressure in engine cylinders, when the point of cut-off is given?

A. No, because that varies with the clearance, the back pressure, etc., as well as with the point of cut-off, etc. Separate tables would have to be made for each clearance, for each initial pressure, and for each back pressure.

Cut off.	Act. Exp. Rate.	Hyp. Log.	Av. Tot. P. Above Vac.	Term. Pres. Above Vac.
.10	7.	1.946	0.2446	0.1428
.125	6.	1.792	0.2990	0.1667
.15	5.25	1.658	0.3487	0.1905
.20	4.2	1.435	0.4370	0.2381
.25	3.5	1.253	0.5133	0.2857
.30	3.	1.099	0.5797	0.3333
.333	2.741	1.007	0.6190	0.3393
.375	2.470	0.903	0.6640	0.4049
.40	2.333	0.846	0.6884	0.4286
.5	1.907	0.645	0.7725	0.5244

The table above gives the actual ex-

pansion rate, and its hyperbolic logarithm, the proportion of average total pressure above vacuum, and absolute terminal pressures, to initial pressure above vacuum, for each of several points of cut off, with 3% clearance at each end (engine non-condensing).

Q. Give full description of how you would proceed to set up a Cowdrey four cylinder engine if every piece was separated from the other.

A. Take the pistons, insert the pitmans in them, and pass the pins into the holes in the piston heads, then place the steam ring on the piston and screw the caps down firmly; place the pistons in the cylinders, holding the topmost one in place, by shoving the pin, that holds the pitman, out a little, so that the pin extends over the top of the cylinder, and then shove the shaft into the inner bearing; then bring the inner ends of the pitmans to their seat on the crank pin, and place the rings around them and bolt them. Then place the lower half of the drum cover in position, and loosely fasten it, then the upper portion, and likewise fasten it, then bolt the

two halves together, and then the whole cover to the drum. Then place the round valve upon the pin that projects into the steam chest, and place the cover on and securely fasten it. If the engine is reversible, place the reverse valve inside the rim or box, screw the lever on, place and bolt the cover, and after making the necessary connections to the boiler, set the oil cups to feeding properly, and the engine is ready to start.

Q. In what position, relative to the main crank, should the valve crank be to work steam full stroke?

A. At a right angle.

Q. How would you increase the lap?

A. By turning the valve pin away from the crank pin.

Q. How can the main, or round valve, be made to properly admit and release steam, if it has any lap?

A. By setting the valve pin so that the outside and inside of the valve expose an equal portion of the opposite ports.

Q. Why is it not necessary to fasten the pins that hold the pitmans in the piston heads?

A. Because the cylinders form guides and contain no openings which they must pass.

Q. One Cowdrey four cylinder engine is equal, in area, to how many engines of the ordinary type with same bore and stroke, and taking steam on both ends ?

A. Two.

Q. Describe the valve, and how it is balanced.

A. A disk with openings near its center, fitted with a steam packing ring, and enclosed on both its faces, and balanced by having steam on its circumference and there only.

Q. Describe the Cowdrey reversing valve, and how it is balanced.

A. A round ring fitted with a steam packing ring, and laid upon its side, and balanced by having steam only on the inside.

Q. State how the reverse valve produces a reverse motion of the Cowdrey engine.

A. By opening the former steam chest to the exhaust, and carrying steam to the inside of the valve, where it finds communication with the opposite cylinder,

whose piston is in mid stroke, checks its return to the end, or outside, of the cylinder, and sends it in the opposite direction.

Q. How would you increase the compression on a Cowdrey engine without altering the lead ?

A. By increasing the thickness of the metal on the inside of the valve.

Q. How would you increase the lead of a Cowdrey engine without altering the compression ?

A. By reducing the circumference of the valve.

Q. What is the Campbell exhaust condenser or exhaust head?

A. This device is intended for use with non-condensing engines as an exhaust-pipe head. It is intended to facilitate the condensation of the steam by providing a condensing chamber with an opening for the admission of air, so placed that the steam, when entering such chamber, will draw cold air in with it, which air, mingling with the steam, will cool it off and also cool the condensing surfaces, whereby the conden sation is greatly increased. Provision is made to collect and carry off the condensed

water without escape through the air inlet, thus depriving the steam of the moisture, and preventing the scattering of water or moisture on roofs and neighborhood, which causes so much damage and often becomes a nuisance. It is stated to relieve back pressure, and to produce great saving of feed-water by returning it. It is claimed that the escaping steam is dry and almost noiseless.

JACKETING.

Q. Suppose that there was just as much condensation in a steam jacket as it prevented in the cylinder, what would be the benefit of the jacket?

A. It would increase the power, or "capacity," of the engine.

Q. What should be done with the "condense" from the steam jacket?

A. It should be returned to the boiler at once, while hot.

Q. Do the pressure and temperature in the steam jacket vary during the stroke?

A. No.

Q. How is the exhaust jacket applied in compound engines in the British navy?

A. In the compound engine at present

so largely employed in the British navy, the cylinders, outwardly, are the same diameter, but the high-pressure cylinder, as previously mentioned, is generally made about ¼ less in area than the condensing cylinder. The space round the actual high-pressure cylinder is used to receive its exhaust steam until the valve of condensing cylinder opens to admit it, as the pistons do not move simultaneously, owing to the cranks being at right angles to each other.

PRESSURES, TEMPERATURES, &c.

Q. In what way is steam "elastic?"

A. It is "elastic" when compressed, not when the volume of the vessel which contains it is increased. But it tends to act as a cushion spring if compressed.

Q. What is meant by steam being "condensible"?

A. It may be rapidly condensed (reduced to the form of water) by the sudden influence of a cooling body as water or a cold metallic surface, or it will slowly return to that state if left to radiate heat to the air through the walls of a

cylinder or pipe without receiving an additional supply of heat.

Q. What is dry steam?

A. Dry steam is that in which there is no unevaporated water, but which is no hotter than saturated steam, that is, no hotter than the boiling point corresponding to the pressure.

Q. What is the difference between a gas and a vapor?

A. A gas requires great pressure and very low temperature to reduce it to a liquid form; a vapor is liquefied at ordinary temperatures.

Q. What then is a vapor?

A. A vapor is any substance in the gaseous condition, at the greatest density consistent with that condition.

Q. Is there an "absolute zero" of temperature?

A. Reasoning fixes an "absolute zero" of temperature at that point where gaseous elasticity disappears=.

—461.2° Fahr.

—274° Centigrade.

—219.2° Reaumur.

Q. What is specific heat?

A. The specific heat of a body is its capacity for heat; the quantity of heat required to raise its sensible temperature 1° F., as compared with the amount required to raise an equal weight of water from 32° F, to 33° F.

Q. What is a "heat unit"?

A. A British "heat unit" is the amount of heat required to raise one lb. of water from 32° F. to 33° F.

Q. Does it not take as much heat to raise one lb. of water from 32° F. to 33° F. as to raise it 1° at any other point of the scale?

A. No. It takes 1¼% more heat, for instance, to raise one lb. of water from 212° F. to 213° F. than from 32° F. to 33° F.

Q. Does the specific heat of other substances than water vary with the temperature?

A. The specific heat of all so called "permanent gases" is practically constant for all temperatures and densities; the capacity for heat of any one gas, is about the same at all temperatures.

Q. Of what use are the "specific heats" of substances to us?

A. By knowing the specific heats of their constituents we may calculate the heating power of fuels, &c.

Q. Of what practical use is it to calculate the heating power of a fuel?

A. By getting an average sample of any fuel analyzed, the amount of steam at 200° F. which a pound thereof should make from water at 212° under theoretically perfect conditions can be calculated; and the amount of air needed, can be ascertained. Then the valve of one fuel found, compared with that of another; and that necessary to perfection of action of the boiler be found out.

Q. What is latent heat?

A. Latent heat is heat which has disappeared or lies hidden. For instance, in evaporating one lb. of water at 212° F. and under a pressure of one atmosphere=14.7 lbs., 966.1 heat units disappear become latent.*

Q. Of what practical use is a knowledge of the latent heat of water?

A. Among other applications I may mention reducing the evaporation-rate of a boiler to a standard from and at 212°.

*Regnault.

Thus having obtained the total heat of evaporation at the actual temperatures of feed water and of steam, dividing that total heat by 966 gives a multiplier by which the weight of water actually evaporated by each pound of fuel is to be multiplied to reduce it to the equivalent evaporation from and at 212°; that is, the weight of water which would have been evaporated by each pound of fuel had the water been both supplied and evaporated at the boiling point, corresponding to the mean atmospheric pressure.

Q. How much heat does it take to generate one pound of steam?

A. That depends on the temperature of the water and the pressure of the steam. The colder the water and the higher the pressure, the more heat required. To make a pound of steam at 212° F. = 100° C., from one pound of water at 32° F. = 0° C., takes three separate lots of heat, as follows: (1) To raise the water from 32° F. = 0° C. to 212° F. = 100 C., it takes 180.9 heat units. (2) To make that water into steam *at no pressure*, 892.9 heat units; and (3) To raise that steam at no pressure to steam

at the atmospheric pressure of about 14.7 lbs. per square inch above vacuum, 72.3 heat units. Total, 1146.1 heat units, equal to 883,789 foot pounds of work.

Q. Are all bodies expanded by heat?

A. Yes, but in different proportions.

Q. How many kinds of expansion are there?

A. Three directions in length, in breadth and in thickness; also three kinds; linear in one direction; superficial, in two directions, and cubical in three directions. Superficial expansion is almost exactly twice the linear expansion; cubical expansion is almost exactly three times the linear expansion.

Q. How much does a hollow body expand?

A. A hollow body expands just as though it were a solid, having the same outside dimensions.

Q. Is the expansion of solids uniform?

A. The expansion of solids between 32° F. and 212° F. is sensibly uniform.

Q. What is the most expansible metal?

A. Zinc.

Q. How much does wrought iron expand?

A. Wrought iron expands about $\frac{1}{7}$ or $\frac{1}{8}$ per cent; (that is, from 1 in 700 to 1 in 800) when heated from 32° to 212° F.

Q. How does heat affect the volume of water?

A. Water on being heated from 30° to 39.1 F. contracts; then expands until at 46° F. it has the same volume as at 32° F.; then expands until 212°, the boiling point under one atmosphere; thence continues to expand as it is heated.

Q. How much does water expand between 32° F. and 212° F.?

A. Between 32° F. and 212° F. water expands .0465; that is, between $\frac{1}{21}$ and $\frac{1}{22}$ its volume at 32° F.

Q. Does the expansion of water take place at the same rate as the increase of temperature?

A. The volume increases in a greater ratio than the temperature.

Q. How can one find the density of water at a given temperature?

A. To find the density of water at a given temperature, (approximately): To the given temperature in degrees F., add 461, and divide the sum by 500. Again, divide

500 by that sum. Add the two quotients and divide 124.85 by their sum.

Thus:—The density of water at 200° F.

$$\frac{200+461}{500} = \frac{661}{500} = 1.325;$$

$$\frac{500}{200+461} = \frac{500}{661} = 0.7564;$$

$$1.325+0.7564=2.0814;$$

$$124.85\div 2.0814=59.98=$$

weight of one cubic foot, at 200°, in pounds.

Q. How can the volume of feed water to produce a given volume of steam, be ascertained?

A. By adding 458 to the steam temperature, multiplying the sum by 373 and dividing the product by the steam pressure in pounds per square inch.

Q. How can we find the admission period requisite for an actual expansion rate?

A. To get the required admission period for a given actual expansion rate—divide the length of stroke, plus clearance, by the actual expansion rate, and deduct the clearance from the quotient.

Thus an engine of 24″ stroke, 5%(=

0.1 foot) clearance, would, to get an actual expansion rate of 4, have to cut off at

$$\frac{2.10}{4} - 1 = .4225.$$

Q. What principle should be carried out in regard to designing and making piston heads?

A. "The piston head should be deep enough to insure tightness, durability and good alignment, without being unduly heavy, especially in vertical engines. It is a good plan to make it of two portions cast from the same pattern, but one having turned on it a hub corresponding to a recess in the other. These two may be bored parallel, smaller than the rod, which is to be turned down parallel to fit the hole accurately, care being taken to leave no square corner at the junction of the full-sized and the reduced portions. It is cheaper to bore parallel than to turn the rod taper and ream a taper hole.

"The ring should be cast larger than the head, three water grooves turned in it, and a sufficient portion cut out diagonally to bring it down to the right diameter when

sprung in to the recess turned for it in the head."—*Power.*

Q. What is the Lancaster & Tonge piston?

A. In this piston, a straight spiral spring, bent into a circle, forces the rings out against the walls of the cylinder, and at the

LANCASTER & TONGE PISTON PACKING.

same time holds them firmly between the junk ring and the piston body. The pressure between the piston rings and the cylinder results from the natural elasticity of the spring and its effort to regain its natural shape, and does not depend upon the screw-

ing down of the junk ring. There is claimed to be no danger of excessive friction being set up between the piston and the cylinder by careless adjustment.

Q. Where should the cross-head pin be?

A. In the center of length of the slides, so as to give straight thrust of these on the guides.

Q. When does an engine "run over?"

A. An engine "runs over" when, if you stand at the cylinder and look towards the fly wheel, the top of the fly wheel runs from you; and *vice versâ*.

Q. How should the crank be fastened to the shaft?

A. It should be both pressed and keyed on.

Q. What is the duty of the fly wheel?

A. The duty of the fly wheel is to store up work during one stroke or portion of a stroke, and give it out to the crank during another period, when the demand is greater.

Q. What class of engines demands the heaviest fly wheels?

A. The more irregular the work, the

heavier the fly wheel must be, to ensure safety to the engine and continuance of the work. The more delicate the work, and the more exacting of regularity of speed, the greater the desirability of a heavy fly wheel, to ensure even quality of the work.

Rolling mills, in which the work is very intermittent and irregular, come under the first head; and paper and flouring mills, cotton spinning factories, and "dynamo" machines for electric lighting, come in the second class.

Q. What rules are there as to the quantity of work to be stored up in the fly-wheel?

A. Watt called for a storage of the work during 7½ strokes, Bourne says 6.

Q. What may be said about centering and balancing the fly-wheel?

A. The fly-wheel must be accurately centered and balanced, especially where it is a pulley-wheel driving smaller pulleys, at higher speeds, in which case its irregularity would be multiplied.

Q. What limit may be assigned for fly-wheel speed?

A. One rule puts 80 feet of rim speed

per second, equals 4800 feet per minute, as the maximum rim speed for a fly-wheel. This would correspond to about 95 turns per minute for a 16 foot wheel.

Q. What is the usual mean radius of the fly-wheel?

A. The usual mean radius of the fly-wheel is from three to five times the crank length (Rankine).

Q. What may be said about fly-wheel pits?

A. These should be provided with good drainage, as wet main-belts are not desirable.

Q. Should fly-wheels be left unguarded?

A. No. They should be caged in to prevent accidents by men or materials being thrown against them, or getting in the pit.

Q. Which need the the heaviest fly-wheels; automatic or throttling engines, and why?

A. Automatic engines need the heaviest fly-wheels, for a given rate of speed and average pressure, because the pressure in the cylinder during one stroke varies more than in throttling engines.

Q. Which needs the heaviest fly-wheels,

an engine which cuts off early, or one which cuts off late; and why?

A. The early cut off takes the heaviest fly-wheel for a given average pressure and given rate of speed, because the cylinder pressures vary more during a stroke.

Q. If two engines have the same piston speed and one has double the number of revolutions the other one has, will they have the same tendency to shake?

A. No. The accelerating force will be twice as much in the former case; and the piston come to its maximum speed in half the time, and in doing it will move over only half the space.

Q. How can an inverted vertical engine be balanced?

A. By using a supplementary steam balance cylinder, above the main cylinder, to annihilate the dead weight.*

Q. How may a marine engine be got off its center quickly and without danger?

A. By having a V-grooved friction wheel on the shaft, and opposite to this placing a short straight block of iron grooved to

* See paper of W. S. Durfee, Cleveland meeting Am. Soc. M. E., 1883.

correspond. This block is connected to the piston of a short stroke steam cylinder and arranged so as to be made to move towards the friction wheel, and touch it, by an eccentric journal.*

Q. What is the leverage of friction of a cylindrical journal?

A. The leverage of friction of a cylindrical journal is the radius of the journal.

Q. How can horizontal stationary engines best be mounted for testing balance?

A. B. W. Payne & Sons, in testing the balance of a horizontal stationary engine, put the engine on rollers, cross-wise with the cylinder axis, and connected with it by a flexible steam pipe. There are few engines that will stand this sort of test, and few men who can bring any single cylinder engine up to the standard imposed thereby, of running without traveling to and fro on the rollers.

Q. How may the weight of a well proportioned engine be approximated?

A. By the cubic foot of cylinder. The following figures are from Haswell:—

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* J. F. Holloway, in a paper before the A. S. M. E., July, 1883.

WEIGHTS OF STEAM ENGINES. Side-wheels.—American Marine (Condensing).

Engine.	Frame.	Water-wheels.	No. of Cylinders.	Volume of Cylinders.	Weight per cubic foot.	Service.
Vertical beam...	Wood.*	Wood.	1	63. cu. ft.	1100 lbs.	River.
do. ...	Wood.*	Wood.	2	216. "	1040† "	Coast.
do. ...	Wood.*	Iron.	1	530. "	1500 "	Sea.
do. ...	Wood.*	Iron.	1	725. "	1089† "	Sea.
Steeple.....	Iron.	Iron.	1	12.8 "	3800 "	River and Coast.
Oscillating.....	Iron.	Iron.	2	540. "	850 "	Sea.
do.	Iron.	Iron.	2	1090. "	760‡ "	Sea.
Inclined direct...	Iron.	Iron.	2	534.5 "	1100 "	U. S. Navy.

* Without frame. † With frame 1109. ‡ Single frame.

English, per Nominal H. P.

Oscillating condensing engines	560 lbs
Wheels	246 "
Boilers	481 "
Coal bunkers	67 "
Water in boilers	294 "

Screw Propellers —Marine (Condensing).

Engine.	No. of Cyl.	Vol. of Cyl.	Weight per cubic foot.	Service.
Vertical direct	2	12.5 cu. ft.	5600 lbs.	Sea.
do.	1	69. "	4660 "	Sea.
do	1	33. "	3650 "	Coast.
Vertical compound, single	2	215. "	2200 "	Sea.
do. double	4	506. "	3010 "	Sea.
Trunk	2	193. "	2080 "	Sea.
Horizontal back-action	2	68. "	4260 "	Sea.

Non-condensing Engine.	No. of Cyl.	Vol. of Cyl.	Weight per cubic foot.	Service.
Horizontal direct	2	3.9 cu. ft.	4990 lbs.	River.
Vertical direct	1	9.3 "	7800 "	Coast.
Inclined direct	2	4. "	5200 "	Coast.

Non-condensing Land-engines.	Engine.	Spur-wheel and connections.	Sugar Mill complete.	Boilers, Grates, etc.	Engine per Cube Foot of cylinder.
Vertical beam, 18 in. × 4 ft.	67,200 lbs.	37,800 lbs.	89,600 lbs.	26,880 lbs.	9600 lbs.
do. 30 in. × 5 ft.	105,000 "	137,159 "	265,879 "	75,000 "	9884 "
Horizontal, 14 in. × 2 ft.	10,914 "	—	—	8,200 "	5100 "
do. 23 in. × 5 ft.	56,000 "	—	—	30,140 "	5600 "

Q. How should bed-plates be leveled up?

A. Bed-plates should be leveled up with sulphur, Portland cement, or lead, their values for this purpose being in the order named.

Q. How can grouting be got in large thin spaces between bed-plates and foundations?

A. By using a stand-pipe through which the cement is introduced.

Q. How may holes be tapped square in cylinder flanges?

A. To tap holes square in cylinder flanges, etc., it is best to have a gauge piece composed of a flat piece of wrought iron having at one end a hub which is bored true square with the flat end, and which has steel bushes or thimbles to fit different taps. A centering plug is employed where desirable, to set the gauge piece over the center of the hole.

WEIGHTS, DENSITIES, FACTORS, &c.

Q. How much does a gallon of pure water weigh?

A. One imperial gallon of pure water at 62° F. has a volume of 277.123 cubic inches and the weight of 10 lbs. But the British

act of 1825, often considered as in force yet, gives it 277.274 cubic inches, and 10.00545 lbs.

Q. How much does one cubic foot of pure water weigh?

A. One cubic foot of water weighs at

32° F= 0°C.	62.418 lbs.
39.1	62.425.
62° (Standard temp.)	62.355.
212° F=100° C.	59.640.

(*D. K. Clark.*)

Q. What is the pressure of a column of water 1 ft. high?

A. A column of water at 62° F. and 1 ft. high exerts a pressure of 0.433 lbs. per square inch on its base.

Q. What height of water column exerts a pressure of 1 lb. per square inch on its base?

A. A column of pure water at 52° F. and 2.3093 ft.=27.71 inches high, exerts a pressure of one pound per square inch on its base. Thus 50 lbs. per square inch would be exerted by a column of pure water $2.3093 \times 50 = 115.465$ ft. high.

Q. How much does 1 cubic foot of sea water weigh?

A. One cubic foot of average sea water

at 62° F. weighs 64 lbs. or 40-39ths, or 1026-1000ths as much as fresh water. 35 cubic ft. of sea water weigh 1 gross ton.

Q. How much solid matter in sea water?

A. Sea water contains 3.40 lbs. of solid matter in 100 lbs., that is, $\frac{1}{30}$th part by weight.

Q. What is the weight of 1 cubic foot of air?

A. One cubic foot of air, under a pressure of 1 atmosphere or 14.7 lbs. per square inch, weighs at 32° F., .080728 lbs=1.29 oz.= 565.1 gr. At 62° F. it weighs .076097 lbs. =1.217 oz.=532.7 gr.

Q. One lb. per sq. inch is how many lbs. per sq. ft.?

A. One lb. per sq. inch is 144 lbs. per sq. foot. It is also 2.0355 inches of mercury at 32° F. or 2.308 ft. of water at 52.3° F.

Q. One lb. of fuel per hour per H. P.= how many foot pounds per lb. of fuel?

A. One lb. of fuel per hour per H. P. is 1,980,000 foot pounds per lb. of fuel, or 2565 units of heat.

Q. 1,000,000 ft. lbs. per lb. of fuel is how many lbs. of fuel per hour per H. P.?

A. 1.98 lbs.

Q. Give the weights and specific gravities of the various materials used about a steam engine.

Materials.	Wt. of 1 cu. ft.	Specific Grav.
Lead sheet	712	11.418
Copper "	547	8.805
Bronze:		
84 cop. 16 tin	534	8.56
81 " 19 "	528	8.46
Brass, cast:		
75 cop. 25 zinc, sheet	527	8.45
66 " 34 "	518	8.30
Steel (least and greatest density)	485 to 493	7.739 to 7.904
Iron, wrought, least and greatest density	466 to 487	7.47 to 7.806
Iron, cast, least and greatest density	378.25 to 467.66	6.9 to 7.5
Tin	462	7.409
Zinc, sheet	449	7.2
" cast	428	6.86
Limestone	178.9	2.58
Granite, gray	180	2.9
Sandstone	130-150	2.08-2.5
Concrete:		
Portland cement 1 to gravel 10	139	2.23
P. cement 1, sand 2	127	2.04
Mortar	109	1.75
Bricks	124.7-135.3	2-2.17
Brickwork	110-115	1.76-1.84
Masonry, rubble	115.3-143.4	1.85-2.3
Sulphur	124.7	2
Coal, Anthracite	85.4-99.1	1.37-1.59
" Bituminous	74.8-81.7	1.2-1.31
" Cannel	74.8	1.2
Lignum vitæ	40.5-82.9	.65-1.33
Oak, American red	54.2	.87
Pine, American	21	0.34
Charcoal (av.) in powder	93.5	1.5
" " in sm. pieces, heaped	25.3	0.405
India rubber, pure	58.	0.93

THE STEPHENSON LINK.*

Q. With the Stephenson link motion, how may the lead and cut-off be equalled?

A. They cannot, with a circular curve in the link, be made the same for both ends of the cylinder, for all points of out-off, and for both front and back gear.

Q. What then is done in this connection?

A. The back gear is disregarded.

Q. Can the lead and the cut-off be equalized for the forward gear alone?

A. No; not for all points of cut-off, with the ordinary circular curve for the link.

Q. What then is done in this connection?

A. The points of cut-off are made the same for ½ gear forwards, because that is the position in which the link is most often used.

Q. How may we change the times of admission for the front and back ends of the cylinder most, in relation to each other?

A. By changing the position of the point of suspension on the links.

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* See also pages 34, 40, of this volume; and page 145 of Part I. For other items on link motion, see pages 34, 35, 37, 41, 44, of this volume.

Q. How may this be done?

A. By moving its point up and down, or vertically.

Q. Where are the links generally suspended?

A. From a point about half way between the eccentric rod connections, and about $\frac{1}{4}$ to $\frac{3}{4}$ inch back of the centre line of the link slot.

Q. Would there be any better point of suspension, as regards steam distribution?

A. Yes, about three inches above the centre.

Q. Why not hang it there, then?

A. Because this would, upon a locomotive at least, necessitate a shorter link hanger; and this would be liable to be too much strained by the motion of the link.

Q. How can you set a Stephenson cut-off so as to cut off at seven feet stroke?

A. I cannot answer your question as to the link motion unless I know the arrangement of parts. If I had that and the dimensions, or rather distance between all centres, I could calculate it for you and give you the answer. Failing that—if I had a rough sketch showing arrangement

of parts, I could give you a rule so you could fill in dimensions and calculate, or perhaps even so you could put the lever right at the point without calculation.

Q. What is the forward or "out" stroke?

A. That made towards the crank.

Q. How can you find the point of exact half-throw of the crank?

A. Square up from the crank centre, at right angles to the guides, or, if the engine be level, "plumb" up from the crank centre.

Q. How can you find the length of the connecting rod?

A. By putting the piston at mid-stroke and measuring from the centre of the cross-head pin to the centre of the shaft.

Q. How do you tell the distance from the centre of the cross-head to the centre of the crank shaft?

A. When the piston is at half stroke, then the distance between these two points equals the length of the connecting rod. At any other point take from the length of the connecting rod the distance of the piston from the inside of the crank end of the cylinder, and add to this half the stroke.

Q. How would you find the length of the eccentric rod of a link motion?

A. Put the valve upon its half stroke, and measure from the centre of the eye to the centre of the shaft; then take off half the shaft diameter, and the thickness of the strap under the foot of the rod.

Q. In an oscillating engine, how can you tell whether the centre of the trunnions and the centre of the crank shaft be fair?

A. Turn the engine over and see that the crank pin brass is always the same distance from the cheek of the crank.

THE ECCENTRIC.*

Q. How may the throw of an eccentric be measured?

A. Subtract the height of the low side from that of the high. Thus:—18″ eccentric, 3″ out of the centre, on a 6″ shaft, has a 3″ low side, and a 9″ high side; 9″ − 3″ = 6″ = total throw, or double the eccentricity.

Q. What will show when the eccentric has slipped?

A. If the engine runs, it will be noted

* See also pages 26, 28, 34, 38, 39, 40, 41, 109, 110, 122, 128, of this volume; and pages 36, 46, 55, 106, 146, 158, of Part I.

that the exhausts from the crank and out ends are not the same.

Q. If the forward-motion eccentric broke, could the engine be worked?

A. Yes; by slinging the backing eccentric in place; but in that case the engine could not be run in the backing motion.

Q. What is a loose eccentric?

A. One having balance weights and stops, used for reversing engines not having regular reversing gear.

Q. How is an engine reversed that has a loose eccentric?

A. The eccentric rods are thrown out of gear, and the slide valve put by hand in mid-travel; then the valve is worked by hand in the direction to reverse the engine; the eccentrics do not turn until the stops get against them; then the eccentric rods are thrown in again.

MEASURING, SQUARING, ETC.

Q. What should be done, when a marine engine is set up, to facilitate setting the cut-off for any given point?

A. It should have prick-punch marks made on the valve stem or some other moving part of the valve gear, showing when

the valve is "line and line" at each end, just about to admit steam; also similar marks as to cut-off, exhaust release, and cushion, where there *is* cushion.

Then the course of the cross-head can be very readily laid down on the guides, and, the piston being held at seven feet stroke, or wherever you wanted to cut off, the valve gear could be put at that notch or clamped at that place, which will bring the "cut-off" witness-mark right for cutting off. The quadrant should then be marked, so that you can get the place the next time without delay or trouble.

Q. What about equalizing the cut-off in marine or other engines having link motion?

A. With most gears, if you cut off at seven feet, or any point on one end, you will cut off earlier or later on the other, unless you sacrifice equality of lead. Generally if you get the cut-offs "squared" for one point of cut-off, the engine will cut off unevenly with any other grade of expansion.

Q. When should the work of marking and setting be done?

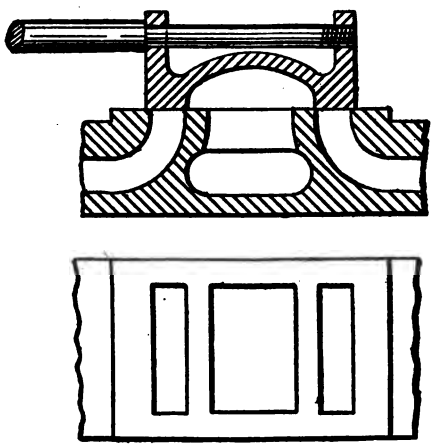
A. While the engine is hot all through;

as an engine set cold will not perform the same when hot.

THE SLIDE VALVE.*

Q. What are the recesses in a valve seat for?

A. Partly to prevent the valve wearing convex so soon, but more especially to pre-



LAPLESS VALVE AND RECESSED SEAT.

*See also pages 23, 31, 32, 108, 110, 111, 118, 122, 131, 166, of this volume; and pages 31, 32, 33, 71, 164, 174, of Part I.

vent a shoulder being worn by the steam edge of the valve.

Q. How far from the steam edges of the ports should they be ?

A. Close enough to the outer edges of the end ports to just let the steam edges of the valves pass them; then there can be no danger of shoulders wearing on the seat.

Q. Can a slide valve be absolutely lapless ?

A. No; there must of course be enough lap to make a steam-tight "seal" at each end when the valve is at mid-position, otherwise the engine would soon "blow through."

Q. Which is the worse, too tight a piston packing or too loose ? and why ?

A. Too tight; because there is danger of cutting the cylinders.

Q. What changes take place in the lead of an inverted vertical engine, as it wears ?

A. The top lead increases, and the bottom lead decreases.

Q. Where a slide valve has variable travel, how is it likely to wear ?

A. The valve is liable to wear convex.

Q. If the amount of lap that an ordinary

slide valve has, be altered, should there be any other change? If so, in what?

A. Yes; the position of the eccentric should be changed.

Q. What should this change be in case some of the lap was cut off?

A. The eccentric should have less angular advance; that is, it should be turned upon the shaft in the opposite direction to its regular forward motion, until the steam edge of the valve has just the desired amount of lineal lead when the crank is upon the centre.

Q. What would be one great objection to a slide valve without lap, outside of the question of economy?

A. Danger of the engine sticking upon the centre, if it were a marine engine.

Q. "How much lap is required for a slide valve having $3\frac{1}{8}$" travel, to cut off at $\frac{1}{4}$ stroke of piston?"

A. If there is no lead, $1\frac{5}{8}$ inch.

Q. Why is it that most types of automatic cut-off engines cannot cut off later than $\frac{1}{4}$ or $\frac{3}{4}$ stroke as a maximum?

A. Where there is a main valve and a riding cut-off valve, the main valve usually has

steam lap proportioned to cut off at some definite point, if the cut-off valve does not effect cut-off sooner; and when the cut-off is by a "release gear," the valve has lap which will cause cut-off at some stated point if the trip does not operate before that time. With most release gears the governor can only cause cut-off while the valve is opening; and this is generally during less than half the stroke.

Q. "Do you recommend inside lap on an engine running at one hundred revolutions or more per minute?"

A. "100 or more turns" means 100 or 1,000, or almost anything, and the piston speed has an influence also. The amount of inside lap depends on too many things to be answered in general. Where there is considerable lead there may be no need of inside lap at even quite high speeds.

The Steam Engine Catechism, Part I., gives considerable information as regards the influence of inside lap, pages 37, 49, 51, 52, 53, 54; concerning lead, pages 41, 43, 51, 55, 125; port area, page 114; clearance, 28, 66, 81, 83.

Q. Can the D form of valve having no

outside lap, employed in automatic engines, work successfully ?

A. The D valve without outside lap cannot cut off nor work successfully unless helped by a riding cut-off valve, or by some other independent cut-off attachment, such as the "Tremper" rig. Of course with a good throttling governor it would work as automatically as any throttling engine; but not economically.

In fact, if the engine was overloaded, a lapless D valve with a good throttling governor would work more automatically than if it had lap, and had to cut off whether or not it was necessary. But it would not, of course, work economically without expansion.

Q. In what case is an Allen valve the most efficient ?

A. Where the valve travel is very short, and the point of cut-off very early.

Q. "Is not more power used up in *moving* double valves, than is saved in steam by the short feed ports ?"

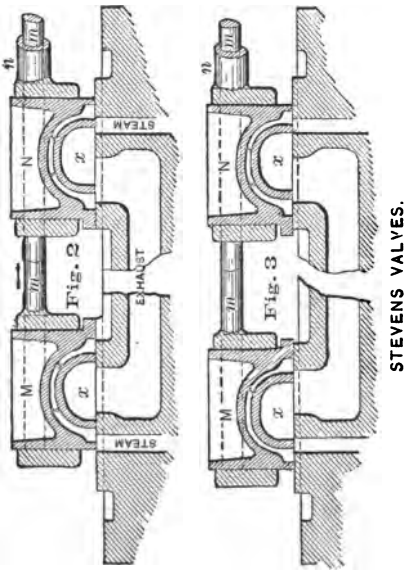
A. The amount of power required to move slide valves, either single or double, is almost an absolutely unknown quantity, and it would be doubly difficult to compare the

amount of steam needed to overcome it, with such a variable quantity as the amount wasted in the clearance spaces, as this last depends on the amount of cushion on other things.

Q. "Why do many engine builders who make double-valve engines, make the exhaust ports so narrow? I contend that the exhaust ports on *all* engines using the D valve, whether single or double, should be twice the width of feed ports, and if a trifle over would do no harm."

A. Most exhaust ports are too narrow, on all classes of engines; double the area of the admission ports would be best for any engine; of course counting the amount of area *uncovered*. If, however, the exhaust ports are large enough to run the engine without back pressure, at highest pressure, latest cut-off and greatest speed, without exhaust release, there is no harm in having the steam ports still larger, particularly where, as with double ported engines (*i.e.*, those having one admission port at each end), the passages are so short that a large cross section of port and passage does not give excessive cushion. Engines with early cut-off need

all the chance that they can get to get the steam in; particularly if they have what is erroneously called "negative lead."



Q. What are the Stevens valves?

A. One variety thereof is very clearly shown in two positions in the annexed cuts,

and the action of the steam edges, both internal and external, may very readily be analyzed by marking a card to correspond with the valve, and moving it to and fro over the seat. In another style the valves have two passages through them; one for admitting and one for exhausting steam to and from the cylinder; the steam is also admitted and exhausted in the usual way. (These valves are upon the Wilson principle.) Another is shown in the cuts here given. This resembles somewhat the Allen valve, with this difference—steam is admitted and exhausted to and from the cylinder through the same passage. This valve also takes steam and exhausts in the usual way, so that the steam is got into the cylinder doubly and exhausted doubly.

Q. A cylinder port is 18 inches long, 2¼ inches wide; the steam lap of the valve is 1 inch, and its travel 6 inches. What is the greatest area that the valve is open for steam?

A. One-half travel, 3 inches; from this take the lap, 1 inch; we have left 2 inches, which is the amount open to steam; this, multiplied by the port length, 18 inches, gives 36 square inches.

Q. How do you find the greatest area that a slide valve is open for steam?

A. Take the steam lap from ½ the actual travel; multiply that by the length of the port.

Q. How can you tell the amount that the exhaust port of a common slide valve is open when the piston is at stroke end?

A. Add the steam lap to the lead; take from this the exhaust lap.

Q. What is the general rule for uncovered port area?

A. Port area uncovered should at various portions of the stroke be proportioned to piston velocity at that portion of the stroke.

Q. What should be the steam port area?

A. D. K. Clark gives as the requisite steam port area, for 600 feet piston speed per minute, $\frac{1}{10}$ the piston area; and for other piston speeds in proportion. This gives us the table on next page of steam port areas for various piston speeds and diameters.

This table can easily be expressed graphically.

Q. How may the various amounts of port opening with the ordinary slide valve in different positions be most easily studied?

Cyl. Diam. inches.	Cyl. Area Sq. inches.	Piston Speed, Feet per Minute.				
		500 .0833	600 .10	700 .11667	800 .13333	1000 .16667
10	78.54	6.55	7.85	9.16	10.47	13.1
12	113.097	9.43	11.31	13.20	15.07	18.85
14	153.938	12.83	15.39	17.96	20.52	25.75
16	201.062	16.76	20.11	23.46	26.81	33.51
18	226.980	18.92	22.70	26.49	30 26	37.83
20	314.16	26.18	31.42	36.65	41.89	52.36
24	452.39	37.69	45.24	52.78	60.32	75.40
30	706.86	58.91	70.69	82.49	94.25	117.81
36	1017.87	84.82	101.79	118.79	135.71	169.65
42	1385.44	115.45	138.54	161.68	184.72	230.91
48	1809.56	150.79	180.96	211.12	241.27	301.60
60	2827.44	235.62	282.74	332.96	376.98	480.12
72	4071.51	339.29	407.15	475.02	542.85	678.60

A. By marking the seat and ports to scale on the top edge of one straight strip of pasteboard or wood, and the valve face on the bottom edge of another strip, marking the centre of each, and on the seat the points of extreme throw of the valve, and then noting how much each port is opened or closed at each position of the valve.

Q. If the length (or width) of the lips of a slide valve is not given, how is it ciphered out from the other dimensions?

A. Add the two laps to the steam port

width. Thus: If a valve has $\frac{7}{8}$ inch outside lap and $\frac{1}{8}$ inch inside lap, and the steam port is $1\frac{1}{4}$ inches, the lips (or "legs," as some call them) are $1\frac{1}{4} + \frac{7}{8} + \frac{1}{8} = 2\frac{1}{4}$ inches each.

Q. How can you get the length of the arch of a slide valve when you cannot get at it to measure it?

A. Add to the width of exhaust port the width of the two bridges, and subtract the width of the two inside laps. Thus: Suppose exhaust port $2\frac{1}{2}$ inches; bridges, each, $1\frac{1}{8}$ inches; inside lap $\frac{1}{8}$ inch at each end; then the arch is $2\frac{1}{2} + 1\frac{1}{8} + 1\frac{1}{8} - \frac{1}{4} = 4\frac{1}{2}$ inches.

Q. Please state the proper proportions used by standard builders for the slide valves of a 19 × 24 passenger locomotive for high speed.

A. I have not at hand many figures for a 19 × 24; but suppose that these, taken from Cooke engines on the D., L. & W. R. R. coal-burning engines, would answer:

Steam ports	$1\frac{1}{4}$" each
Bridges	$1\frac{1}{8}$" "
Exhaust port	$2\frac{1}{2}$"
Valve arch	$4\frac{1}{2}$"
Steam lap	$\frac{7}{8}$" each end.
Exhaust lap	none.

I would recommend $\frac{1}{32}$ inch exhaust lap

each side, which would bring the valve arch down to $4\frac{1}{16}$ inches. I would make the ports at least 18″ long.

Q. What should be the travel of a lapless valve?

A. A lapless valve should have for travel double the steam port width.

Q. What should be the travel of a lapped valve?

A. A lapped valve should have for travel at least twice the sum of the widths of leg and of steam port.

Q. What effect on the cut-off has increasing the travel?

A. Increasing the travel very slightly retards the cut-off.

Q. What is "over-travelling"?

A. Over-travelling is where the steam edge of the valve passes the end port and travels over on to the bar or bridge.

Q. What is the effect of having so little travel as to open the steam port only part way?

A. If the travel is so short as to only partially uncover the steam port, the steam is apt to be wire-drawn, at least at high piston speeds.

Q. Does a plain D slide valve move with the piston of a steam engine, or against its motion?

A. It does both.

Q. When the piston is at the back end of the cylinder, where is the valve, supposing that there is no rocker arm and that the engine has not negative lead?

A. It has moved quite a little distance forward, so that the back steam edge of the valve is just uncovering the back edge of the back steam port.

Q. When the piston has got well toward the centre of its stroke, what is the valve's action?

A. It commences to move in the opposite direction to the piston.

Q. When the piston is very nearly at front stroke end, where is the valve?

A. It has commenced to open the front port for the purpose of admitting steam to the front of the piston.

Q. Up to the end of the stroke, how does the valve run?

A. In the opposite direction to the piston.

Q. When the piston reverses to go on the back stroke, what is the valve movement?

A. They travel in the same direction (backwards) until the valve has made its full back stroke.

Q. What then?

A. The valve reverses and goes forward, running forward until the piston is at back stroke (engine on back centre).

Q. What next?

A. The piston reverses and runs the same way as the valve, as on the preceding forward stroke.

Q. Lay this down for a 12" stroke engine.

A.

PISTON RUNNING FORWARD.	VALVE RUNNING.
Distance travelled 0"	Forward.
3	(Still.)
6	Backward.
9	"
12	"

PISTON RUNNING BACKWARD.	
Distance travelled 0"	Backward.
3	(Still.)
6	Forward.
9	"
12	"

Q. How may you note the actual and relative motions of piston and valve?

A. By watching the rods.

Q. Why should every engineer do this?

A. Because some motions open the valve suddenly and close it suddenly, and some open and close too leisurely.

Q. Does the steam chest always contain live steam?

A. The steam chest in the "Buckeye" engine contains exhaust steam only.

Q. Is an absolutely balanced valve desirable?

A. There are many who claim that an absolutely balanced valve would not be desirable, because enough friction to keep the surfaces bright and prevent corrosion from starting leakage is necessary.

Q. What is the rule ordinarily given for the pressure which tends to force a slide valve to its seat?

A. Add the area of one steam port to the area of the valve port, and multiply these by the mean effective pressure per square inch.

Q. Does the pressure in the steam chest (tending to press an unbalanced slide valve to its seat) vary during the stroke? If so, why?

A. "Steam chest diagrams," taken by the

indicator, show that where the chest is small and the supply pipes insufficient in carrying capacity, the pressure in the chest, tending to press the valve to its seat, is least during the period when the cylinder is taking steam; say, during about the first and fifth half quadrants of crank position of the average automatic cut-off engine, cutting off at one-fifth to one-fourth stroke when properly loaded.

Q. Does the pressure under a slide valve (tending to lift a slide valve from its seat, if it is free to move) vary during the piston stroke? If so, why?

A. The pressure under a slide valve, tending to lift it from its seat, being simply that exerted upon it through the end ports and exhaust ports, of course varies greatly during the stroke. Just what are these upward pressures per square inch, at various positions of piston, is easy to determine; but what are the valve areas thus exposed to upward pressure, is much more difficult to analyze, owing to the fact that the valve is shifting, and that its speed and direction, compared with those of the piston, are constantly changing.

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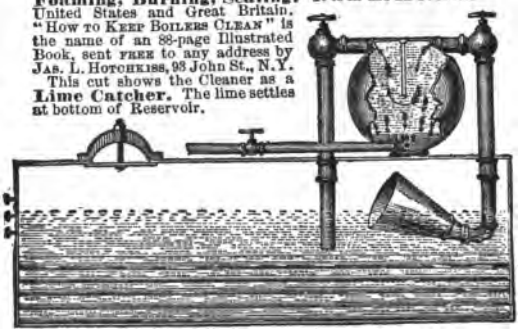

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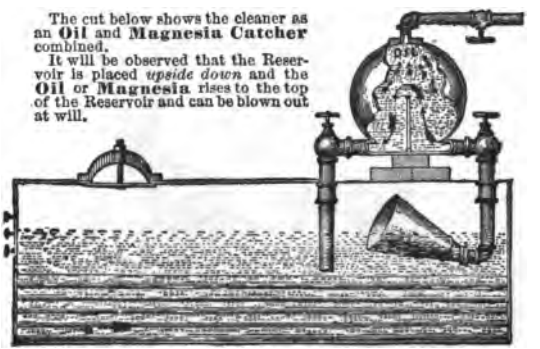

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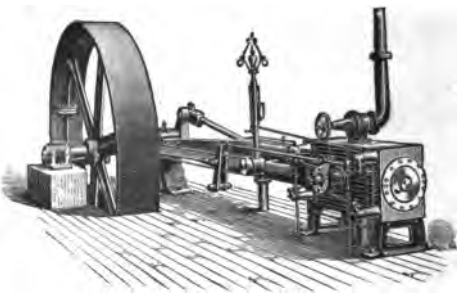

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